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UNCLE SAM'S GRAVEYARD OF DEPARTED HOPES.

BY REAR ADMIRAL ELLIOT SNOW, CONSTRUCTION CORPS,
U. S. NAVY, MEMBER.

Our forefathers when framing the Constitution of the United States included a clause which empowered Congress "to promote the progress of science and useful arts by securing for a limited time * * * to inventors the exclusive right to their * * * discoveries." Whereupon in 1790 a law was enacted to grant "letters patent," as they were termed, to inventors and placed the duty of granting these rights, on the Secretary of State, the Secretary of War and the Attorney General or any two of them (there was no Secretary of the Navy at that time). Now administrative conditions are entirely different and our government has under the Treasury Department a Commission of Patents to attend to such matters.

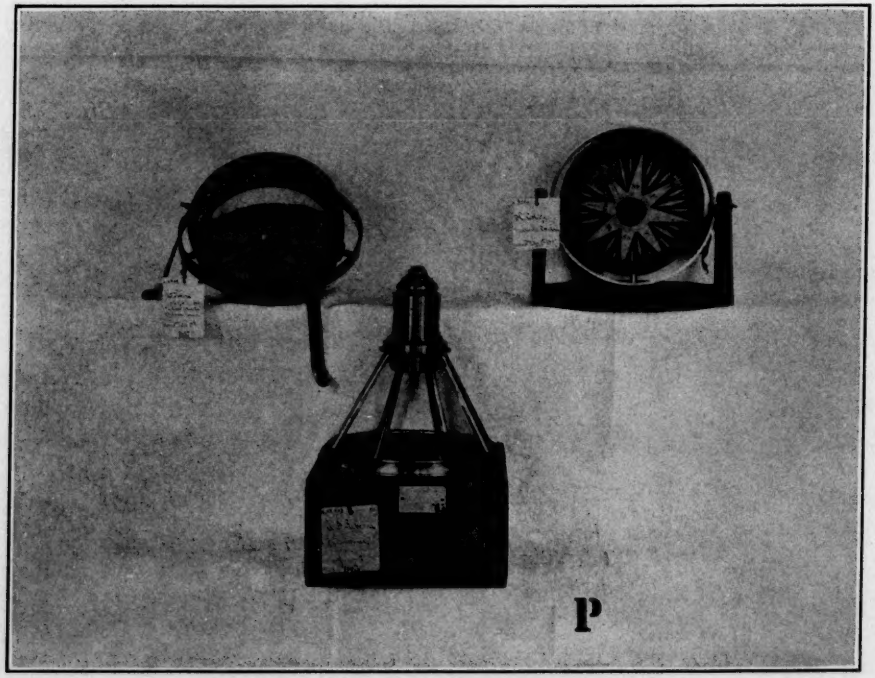
Of the useful arts, which have been promoted by the progress of science that of shipbuilding has gone ahead by leaps and bounds since the first frigate was built by the United States—the one

built for tribute to the Algeine Corsair Hasan Bashaw—the *Crescent* of 36 guns. This progress has been due to the stimulation that American inventive genius has received through stress of war, the War of 1798, with France; of 1812 with England; of 1846 with Mexico; of 1860-65 between the North and South; of 1898 with Spain; and of 1917 with Germany and because of the fierce commercial competition on the high seas, when our hardy shipmasters of the merchant marine sought to grasp the golden opportunities in the West and East Indies and the Orient.

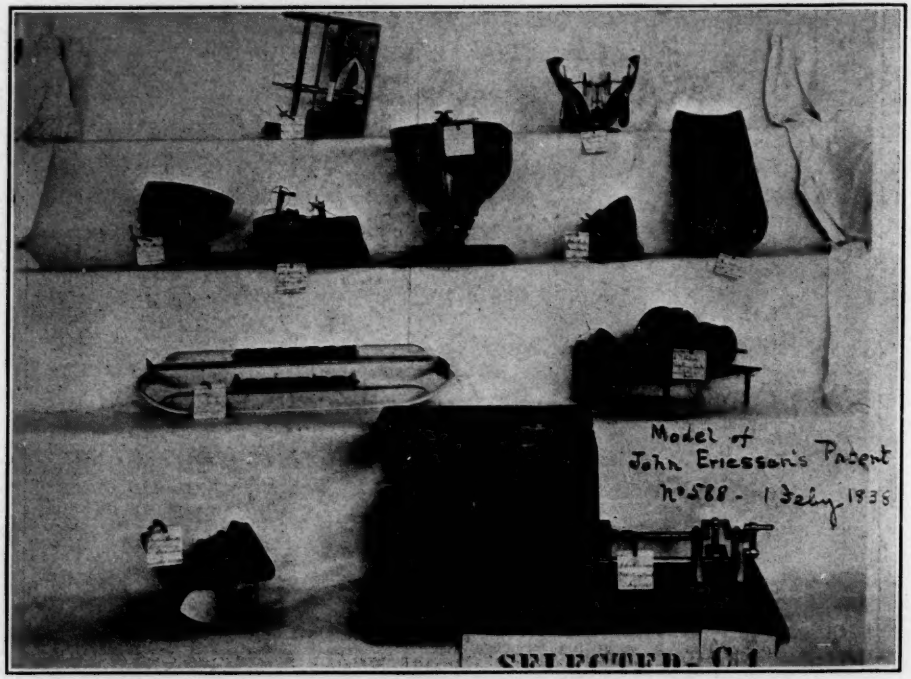
During the Civil War so many ideas connected with implements of war and details of ship design and construction poured into Washington, intended to help save the Union that President Lincoln appointed a special board or committee to pass judgment on the merits of the ideas and inventions thus presented. Today, this has grown into the National Academy of Science in Washington, D. C. During the World War more than one hundred thousand schemes of one sort or another were passed upon by technical committees whose work was coordinated by the "Aide for Inventions" under the Secretary of the Navy. Some, but relatively few of these ideas, schemes, and inventions—the result of war stimulated imagination—were of value. Many were made the subject of "letters patent." In these the instrument or device was described in detail and sometimes was accompanied by a small working model, of a greater or lesser degree of precision of workmanship, but in the main very crude. Since 1886, the Patent Office has not required any models to be deposited with it as a requisite to granting a patent.

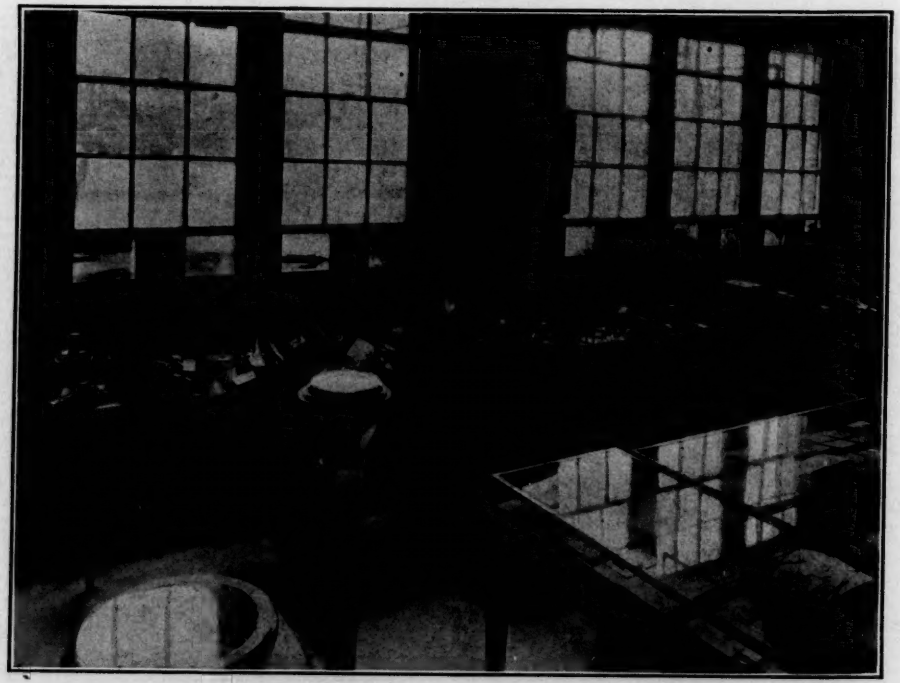
The great majority of models, deposited by inventors ever since the Patent Office was instituted, were destined through inherent defects in the conception of the idea they embodied, never to pass beyond the "letter patent" stage. Thus Uncle Sam's Graveyard was opened and gradually became filled to overflowing with monuments of abortive ideas and abandoned hopes. The collection grew and grew until some step had to be taken to clear away and bury forever from official sight and memory, these tangible evidences of the "fertile imaginative genius of America."

The disposition, removal and distribution by sale or otherwise of all models, deposited with the Patent Office prior to 1886, was made

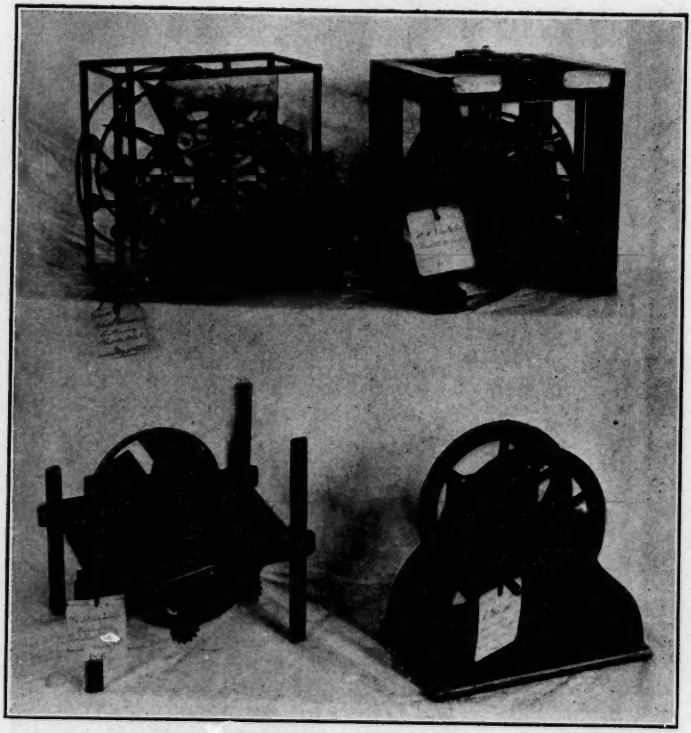


THE CELEBRATED RITCHIE COMPASSES.


Model of
John Ericsson's Patent
No 588 - 1 July 1838



PATENT OFFICE MODELS IN NAVAL EXHIBIT SESQUI CENTENNIAL 1926.



FEATHERING PADDLE WHEELS OF BIZARRE CONCEPTION.

possible by an Act of Congress, approved February 13, 1925. The act required that this clearing out be completed on or before June 30, 1926, and to make this possible the sum of $10,000 was placed at the disposition of the Commissioner of Patents. The terms of the act were announced in the "Official Gazette of the United States Patent Office" on June 16, 1925.

As soon as this authority was granted to the Commissioner of Patents and the funds placed at his disposal became available the despoiling of Uncle Sam's Graveyard of Departed Hopes was taken up in earnest. For a long while the models had been boxed up (there were nearly 800 large boxes all told) and stored in an old garage or storeroom back of Hayworth's Printing Company in Washington, D. C., on G Street between 6th and 7th Streets, N. W. The "remains,"—and shall it be added these followed by the ghosts of inventors long since dead and buried with their hopes—were carted away in rough dead boxes down an ill-paved alleyway to the eastward of the building which had so long been their home.

All models did not however receive such an indecorous handling. It was only the unclaimed "remains" which were thus carted away to a potters field. The act provided that other disposition could and should be made of some of them. Priority of claim rested with the inventors who had deposited the models or with their heirs, assigns or representatives; next came Federal, State or private museums or institutions. Under this provision, many models found their way back into the hands of nation-wide-known inventors like Thomas A. Edison and Henry Ford. Others went to the National Museum (Smithsonian Institution); others found a permanent home in the Bureau of Efficiency and the National Guard of the District of Columbia. Still others pertaining to the Art of Shipbuilding passed under the scrutiny and into the custody of the Bureau of Construction and Repair, Navy Department, Washington, D. C.

Great was the variety of models pertaining to the Art of Shipbuilding, but greater still was the variety in other useful arts. A contemporary newspaper account listed the original Mason fruit jar; the peanut roaster; artificial legs; devices for walking on water; street car fare boxes dating back to the Civil War; false

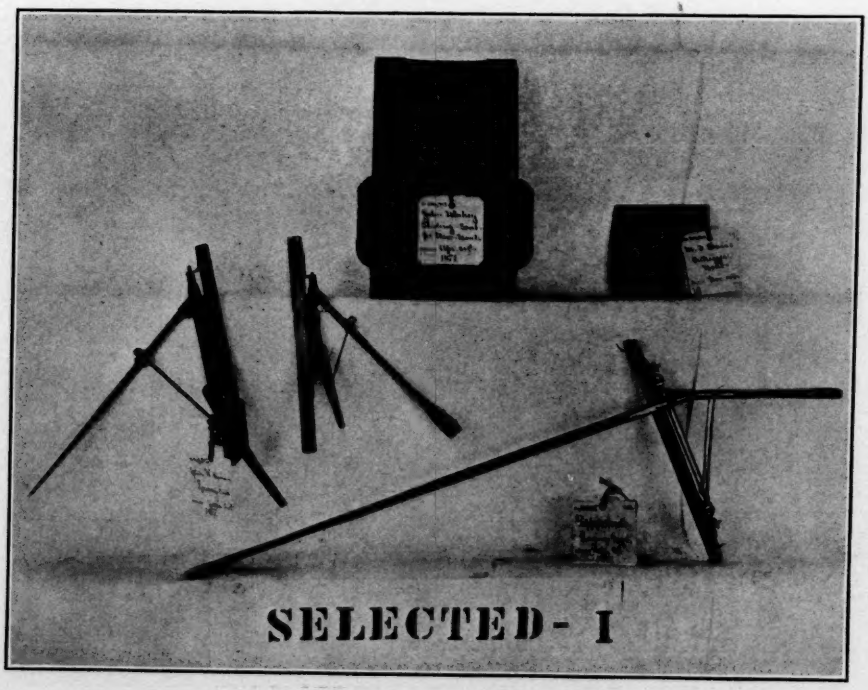
teeth; a baseball bat with a spring in it to give increased range of drive for a home run; a shoe shining machine that rang a bell and stopped when the shine was completed, and what not? Summing them all up they totaled over 75,000. The discarded ones created a musty pathetic atmosphere of their own as they lay, strewn about wings Nos. 3 and 4 on the third floor of the General Supply Committees building of the Treasury Department, awaiting an interested bidder. The first large lot brought $1,500.00; the next was withdrawn because only $900.00 was bid. If these could have been sold piece meal what a glorious chance youngsters would have had to acquire interesting but quite useless "gadgets."

The models pertaining to the Art of Shipbuilding, as they were scrutinized were carefully listed, and examined with the view to retaining such as might be of particular interest from a technical historical standpoint. One of the models found and preserved was a ship propelling device for making two propellers, arranged in tandem, revolve in opposite directions—as do today the propellers of our torpedoes. It was John Ericsson's Patent No. 588, applied for from London while Ericsson was still a citizen of Sweden. The model was on view in the Bureau of Construction and Repair, at the time the Crown Prince of Sweden visited Washington to unveil the statue of John Ericsson.

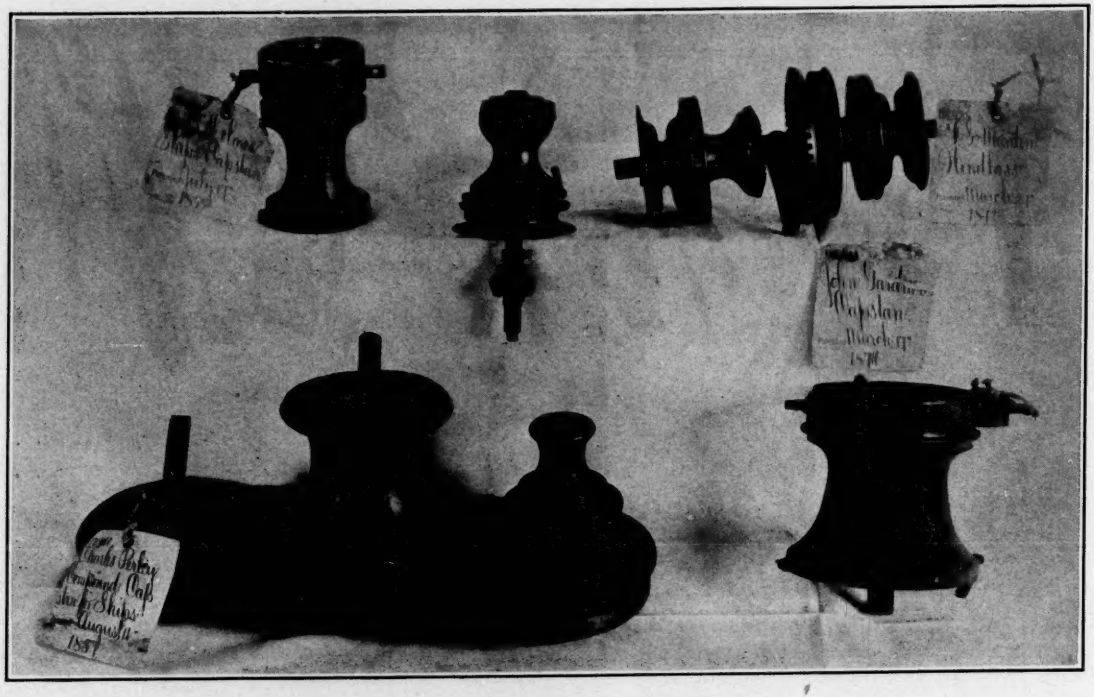
As the examination of the models pertaining to ship construction proceeded it became evident that some method of classification would have to be adopted and for record purposes the models would have to be photographed in groups. The listing of the models recorded the name of the inventor, date and number of the invention and its general classification.

Of the several large lots thus examined some of which included several hundred models, the following is a typical lot:

Device	*No. of models*	*Range of dates*
Ship Construction	121	1844-1895
Paddle Wheels	46	1838-1881
Screw Propellers	56	1838-1897
Vibrating Propellers	38	1841-1880
Wave Propulsion Devices	3	1867-1879



Oddly Contrived Rowing Devices.



SHIP WINDLASSES AND CAPSTANS.

Steering Apparatus	79	1842-1880
Armor	21	1861-1878
Torpedo Boats	2	1876-1877
Ordnance	31	1854-1877
*Life Preserving Appliances	58	1837-1888
Anchors	50	1848-1883
*Rowlocks	25	1867-1884
Rigging Devices	11	1857-1884
Reefing Apparatus	5	1855-1858
Booms, Boom Irons and Sheaves	12	1855-1879
Rope Clamps	10	1868-1879
Salvage Apparatus	22	1841-1879
Cable Stoppers	7	1856-1875
Binnacles and Compasses	3	1847-1865
Berths	13	1841-1880
†Miscellaneous	52	1839-1881

As the examination of these models was under way at the time of the Sesqui Centennial Exhibition in Philadelphia, it was decided to include a few of them in the Naval Exhibit. The display was an interesting one to all whose interest lies along the line of technical history. Unfortunately the limited space allotted crowded them to such an extent as to render the display considerably less effective than it might otherwise have been. With the closing of the Naval Exhibit and the distribution of its objects, the Patent Office models from Uncle Sam's Graveyard of Departed Hopes, along with the other objects are headed for oblivion.

Specifications of Letters Patent No. 588, John Ericsson's "Propelling Steam Vessels," with accompanying drawing are reproduced in full.

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* Models were not required after 1886.

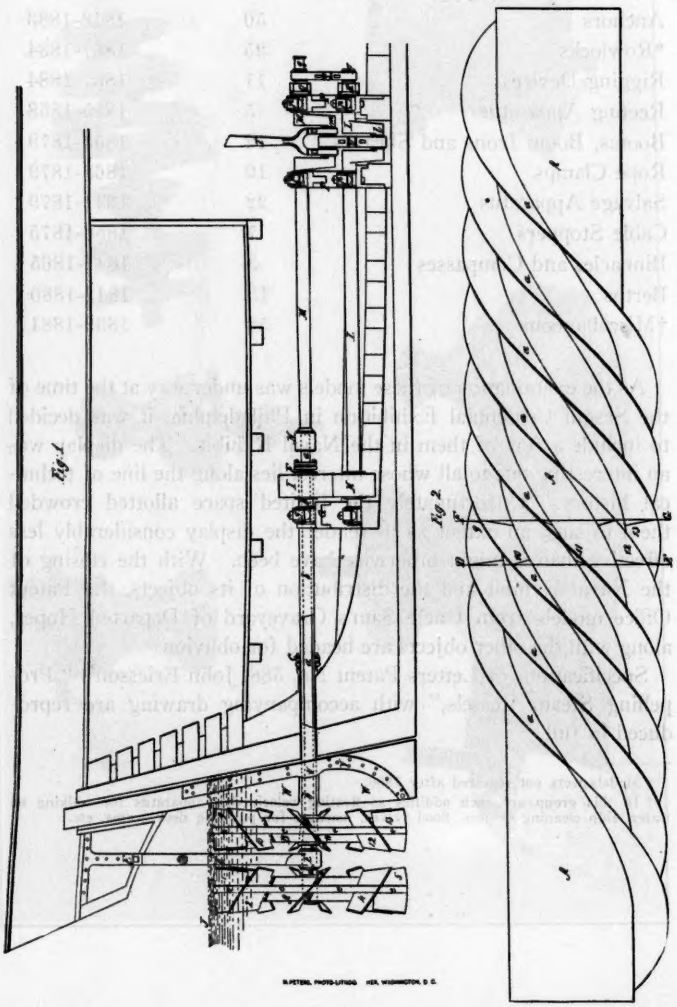
† In this group are such oddities as floating velocipedes, apparatus for walking on water, ship cleaning devices, flood valves, machine for puttying deck seams, etc.

Sheet 1, 2 Sheets.

J. Ericsson

Screw Propeller

No. 588. Patented Feb. 1, 1838.



B. PETERS, PHOTO-LITHO. WASHINGTON, D. C.

Sheet 2, 2 Sheets

J. Ericsson.

Screw Propeller.

No 588. Patented Feb 1, 1838.

Fig 3

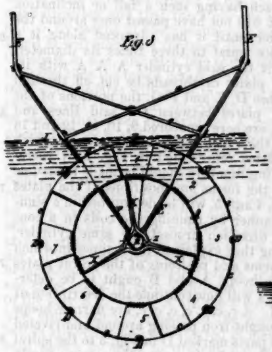


Fig 4

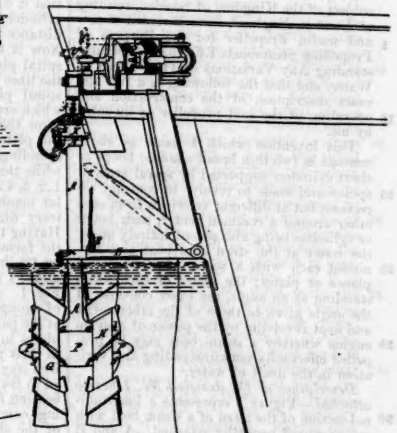


Fig 5

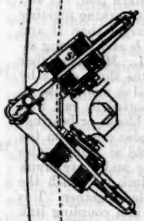
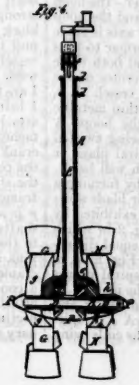
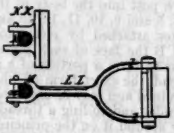


Fig 4





UNITED STATES PATENT OFFICE.

JOHN ERICSSON, OF LONDON, ENGLAND.

PROPELLING STEAM VESSELS.

Specification of Letters Patent No. 588, dated February 1, 1838.

To all whom it may concern:

Be it known that I, JOHN ERICSSON, a subject of the Kingdom of Sweden, residing at London, England, have invented a new and useful Propeller for the Purpose of Propelling Steamboats Effectually Notwithstanding Any Variations in Their Draft of Water, and that the following is a full and exact description of the construction and operation of the said propeller as invented by me.

This invention which I name as above consists in two thin broad metallic hoops or short cylinders supported by spiral arms or spokes and made to revolve in contrary directions but at different velocities from each other around a common center, such hoops or cylinders being also placed entirely under the water at the stern of a boat and furnished each with a series of short spiral planes or plates; the plates of each series standing at an angle, the exact converse of the angle given to those of the other series and kept revolving by the power of a steam engine whereby a steam boat may be propelled effectually notwithstanding any variation in the draft of water.

Description of the drawing No. 1 hereto attached.—Figure 1 represents a longitudinal section of the stern of a steam boat with my improved propeller attached. A and B are two cylinders or broad hoops of wrought iron supported by spiral arms or spokes which will be explained hereafter—the hoop A is attached to the axis *a a a* and the hoop B to the axis *b*, *b*, *b*, which latter axis is made hollow in order to admit the former to pass through and work within it and both these axes pass directly through the center of the stern post into the body of the vessel. 1, 2, 3, 4, 5 and 9, 10, 11, 12, 13 are thin metallic plates attached by rivets to the hoops A and B the face of each plate being twisted so as to form a portion of a spiral plane or thread, the exact form of which will be determined and may be obtained by forming a cylinder and coiling a thread or blade spirally around it on the principle exhibited by the diagram represented in Fig. 2 in which A A A represent a cylinder of equal diameter with the hoops A and B in Fig. 1. *a*, *a*, *a*, *a*, *a*, *a*, *a*, are eight thin spiral planes or plates of the same width as the plates 1, 2, 3, 4, 5 and 9, 10, 11, 12 and 13 in Fig. 1, and coiled around the said cylinder A A A spirally like the thread of a screw the coils being placed at equal distances from each other and each having such a fall or inclination, that it will not have passed once around the cylinder until it has advanced along it a distance equal to three times its diameter. Now if the said cylinder A A A with its spiral plates or threads be cut off through the lines D, E and F G, the portions of the spiral plates between the said lines and which are here numbered 9, 10, 11, 12 and 13 show the exact forms and positions which the plates represented in Fig. 1 by corresponding numbers should be made to assume while the forms and positions of the plates 1, 2, 3, 4 and 5, will be determined in a similar manner by running the coils in a contrary direction around the same cylinder. Having thus explained the manner in which the forms and positoins of the spiral plates on the hoops A and B ought to be determined I will now continue the description of Fig. 1. *c*, *c*, *c*, *c*, *c*, *c*, *c*, *c*, are narrow hoops of wrought iron passing around and riveted at the parts marked D in Fig. 3 to the spiral plates in order to secure them more firmly in their places. E E E is a strong wrought iron stay (better seen in Fig. 3) firmly bolted to the stern of the vessel. *e*, *e*, is a brass bearing fixed in the said stay E which bearing carries the outer and enlarged end of the shaft *a*, *a*, *a*, the other end of the shaft being carried by and working through a stuffing box F attached to the shaft *b*, *b*, *b*, which shaft is supported by a strong cast iron framing G and plumber block *g*. C is a stuffing box fixed to the stern post to prevent the water from entering the vessel around the shaft *b*, *b*, *b*, which should work freely through the stern post. H and I are two broad cog wheels working together I being about one fifth larger than H and attached to the shaft *b*, *b*, *b*, and H being attached to a crank shaft L L. M M is another crank shaft attached to the shaft *a a a* by the coupling box N. *l* and *m* are cranks on the shafts L and M supported by cast iron frames P P and plumber blocks or bearings *p p p p*. Q and R are also cranks on the shafts M M and L L fixed at right angles to the cranks *l* and *m*. *q* and *r* are crank pins and S a coupling link by which the cranks Q and R are coupled together. T is a connecting rod and U is a coupling link attached to the cranks *l* and *m*. This connecting rod is to be connected in the ordinary manner to the piston rod or beam of a

steam engine the cylinder of which may be placed either vertically or horizontally across the vessel. Another connecting rod connected to another engine may be attached to the crank pin *q* in a similar manner by which a more regular power will be communicated to the cranks and shafts *m* M and *l* L. It is evident that if motion be communicated to the cranks *l* and *m* the shafts L and M must be turned around in one and the same direction and that therefore the shaft *b*, *b*, *b*, by means of the unequal cog wheels I and H will move in a contrary direction to the shaft *a*, *a*, *a*, and at a less speed and at the same time the broad hoops A and B with their spiral plates will move in contrary directions and at unequal velocities; it should be stated that when the cylinder A and its plates 1, 2, 3, 4 and 5 as viewed from the vessel revolve to the left the vessel will be propelled forward and when moved to the right the vessel will be backed. W W is the rudder divided into two parts held together by two strong wrought iron stays V fixed one on each side having wide loops or bends at *v* to admit of the free motion of the rudder. Fig. 3 represents an end view of the broad hoops A with its spiral plates. *e e e e* are cross stays to give additional strength to the stay E E. X X X are the 3 wrought iron spiral arms or spokes to the hoop A before alluded to and constructed in manner here shown in order to prevent the resistance which would otherwise be presented by them to the progress of the vessel. These arms all meet in the center where they are welded to a boss Y which is afterward bored to receive the shaft *a*, *a*, *a*, upon which it is firmly keyed by the keys *z z z*. D D D D D D D D are the angle pieces which join the several pieces C C C C C C C C C of the narrow hoop that supports the spiral plates on the broad hoop. J represents the ordinary water line.

And I, the said JOHN ERICSSON, do hereby declare that the application of my propeller represented in the annexed drawing No. 2 and hereafter described will be highly useful either for ships of war or merchant vessels.

Description of the drawing No. 2.—Fig. 4 represents a longitudinal section and Fig. 5 the plan of the stern of a vessel with my propeller attached and Fig. 6 is a section showing the manner in which the requisite contrary movement is obtained in the said application of my propeller. In order the more clearly to describe the said application I will first describe this last mentioned Fig. 6, but previous to doing I have to state that similar letters of reference will be used to denote similar parts in all the figures. A is a hollow stem of wrought iron, to which are welded collars *d d* and broad flat branches or arm *a a*. B is an axle or shaft of steel on which a hollow axle C of wrought iron is made to work, both these axles B and C are supported by the flat arms *a a* which arms have eyes or bearings at D D in which the respective axles work. E is an axle or upright shaft of wrought iron working through the hollow stem, A. F is a crank attached to the said upright shaft by a sliding coupling box *f* and key *x*. *b*, *c*, *e* are conical cog wheels working together and firmly fixed on their respective axles B, C, E by which it becomes evident that if motion be given to the upright shaft the axles B and C, will move in contrary directions. G, H, are two thin broad hoops of wrought iron, G being firmly fixed on the axle B and H fixed on the hollow axle C. Each hoop is provided with a series of spiral planes as described in the foregoing description of my improved propeller applicable to steam navigation. *g g* and *h h* are the spiral spokes also before described but which spokes in addition to their twisting or spiral form are here curved or bent outward in order to give room for the flat arms *a a* of the hollow stem. In order to protect the conical wheels as well as diminishing the friction which they would produce in passing through the water a drum P P P of slight metal divided into three parts and pointed toward the ends, is made to inclose all the gear work under water, the central part of which drum, being fixed to the arms *a a* and the pointed ends or caps fixed to the spokes of the propellers slits being made for that purpose and a space of about one eighth of an inch left between the three parts in order to admit of a free and contrary movement of the propellers. Having thus described Fig. 6, representing the section of my propeller as applied to ships of war or merchant vessels, I will now proceed to describe Fig. 4, showing the manner in which it is attached. K is a bracket of iron (better seen at K K), it is firmly fixed to the stern for the purpose of carrying the propeller, by means of the hollow stem A and its collars *d d* the hollow stem being kept in its place by the key *k* which is secured to the bracket K by a slight chain. L is a stay of wrought iron, to keep the hollow stem A firmly in an upright position; and to receive and communicate the force of the propellers for which purpose it is attached to the stern post by hinges on each side of the rudder; its form will be better seen at L L, the rudder being made to work between *l*, *l* and the fork M fitted to receive the hollow stem A, which is kept in by a key *m* this being secured to the stay by a slight chain as shown in the drawing. N is a ring or collar around the upper part of the hollow stem A, having a strong loop or eye at *n*. It is evident that by driving out the keys *m k* and *x* and pushing down

the sliding coupling box *f* the hollow stem with the upright shaft and propeller will become quite detached and may thus be lifted out of the water or taken on board simply by applying a hoisting tackle at *n*. The stay L should whenever the propeller is taken on board be lifted up and kept suspended by the hooked rope at R in the position shown by the dotted lines *r r*.

I have now to state that the most advantageous mode of giving motion to the propeller is that of applying a steam engine to the crank F, and I would recommend an engine so arranged as represented in Fig. 5, S S being two high pressure steam cylinders placed horizontally and nearly at right angles, their power to be communicated by the forked connecting rods *s s* to the crank F. In cases where the application of steam engines would be objectionable manual force may be applied by means of long winches similar to those used for working ordinary chain pumps or by means of a capstan made to give motion to the conical cog wheel T, (marked in red lines, see Fig. 4,) such cog wheel to work in or give motion to another conical cog wheel V fixed on the upright shaft, F.

Now whereas the use of spiral planes acting obliquely against the water and moving in contrary directions for propelling steam boats is not new, I do not claim as my invention the use of such spiral planes or their contrary motion; but

I claim as my invention—

1. The metallic hoops or cylinders and the spiral arms or spokes hereinbefore described together with the entire immersion of the propeller by which means I am enabled to employ the whole surface of all the spiral plates at one time and whereby the beneficial result of a great propelling force will be obtained by a propeller of much less dimensions than heretofore.

2. And I also claim as my invention the giving a greater speed to the outer series of spiral plates which move in the current produced by the motion of the other series and by which greater speed the beneficial result of saving of power and increased propelling force will be obtained.

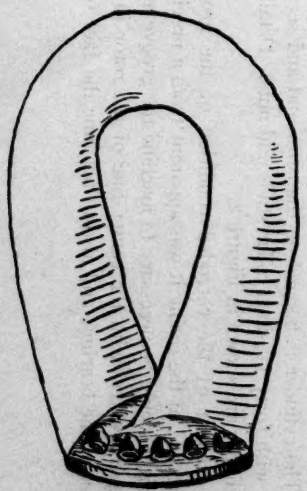
3. And I further claim as my invention the application of the propeller as described in drawing No. 2—that is to say: 1stly, I claim the upright hollow stem with its arms or branches for carrying the propeller by means of which stem the propeller may be either suspended and immersed under the water when required to be used, or on other occasions lifted out of the water so as not to interfere with the sailing of the vessel; 2ndly, I claim the drum or conical casing for protecting the bevel wheels and for diminishing the resistance in passing through the water; 3rdly, I claim the attaching the propeller to or detaching it from the engine or other power employed on board the vessel by means of a coupling box at the upper end of the upright shaft of the bevel wheels.

J. ERICSSON.

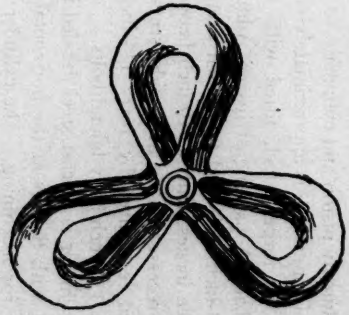
Witnesses:
JAMES M. CURLEY,
JOSEPH MARQUETTE.

Myers.

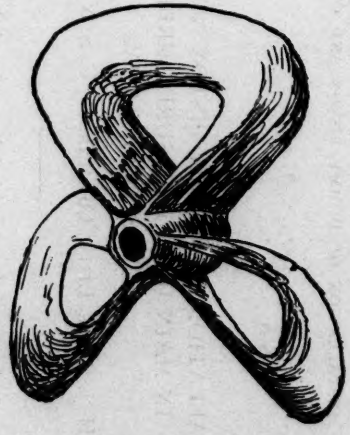
(January 23rd 1896.) "Patent Propeller."



The Blade.



Ordinary Propeller.



Towing Propeller.

NOTE: The above figures showing Myers "Patent Propeller" although not connected with Captain Snow's paper are included here as of possible interest.

INSTALLATION OF DUMMY PISTON AND CYLINDER IN MAIN HIGH PRESSURE TURBINE OF U. S. S. *LEVIATHAN.*

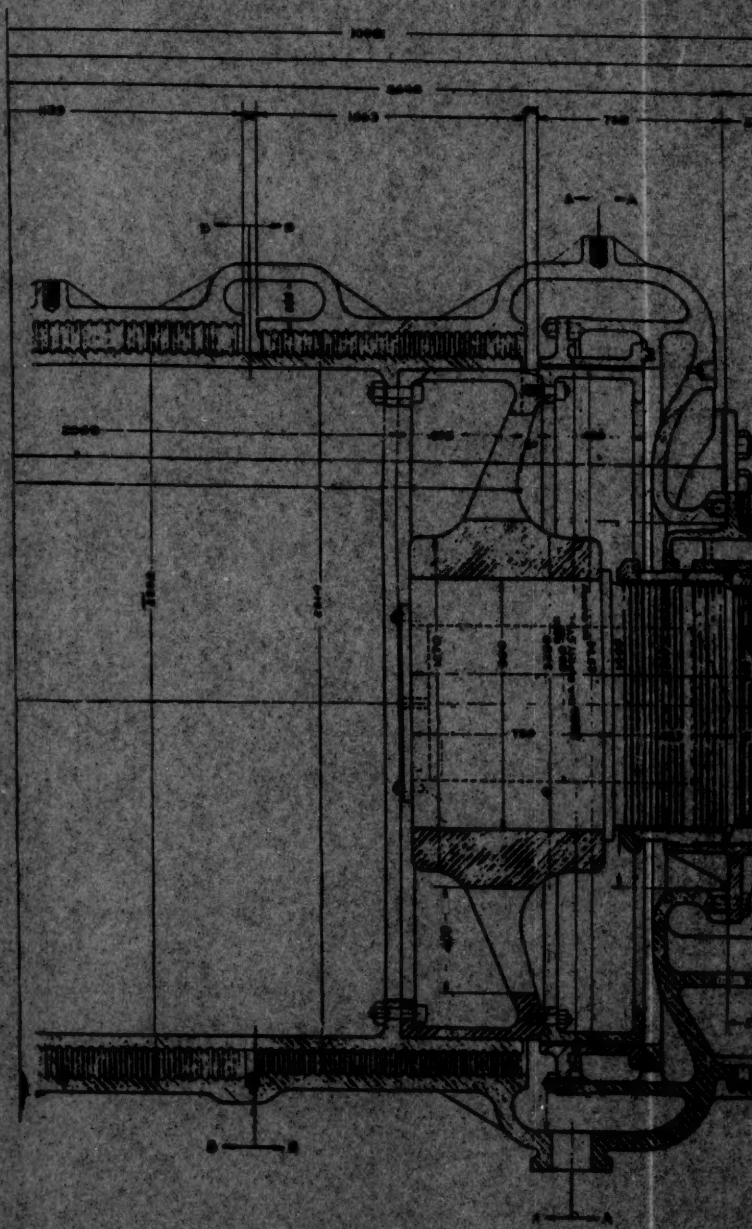
By H. J. Reuse, Lieut. Commander, U. S. Navy, Member.

One of the most interesting jobs accomplished by the Navy Yard, New York, during the winter overhaul of the U. S. S. *Leviathan* was the installation of a new dummy piston and cylinder in the main high pressure ahead turbine. Figure 1 is a sectional view of the forward end of the turbine showing the location of the dummy piston and cylinder.

The dummy piston, which is approximately eight feet, six inches in diameter, is of forged steel, fitted with seventeen rows of radial, straight-backed, packing strips of Admiralty metal, rolled in grooves 3/16-inch wide and 9/32-inch deep in lengths of approximately three inches, with 0.010-inch between the butts. The piston, which is recessed on the after side, seats on a spigot on the rotor wheel, and is secured by forty 1½-inch collar studs and nuts, and twenty 1¼-inch body bound bolts.

The dummy cylinder is of cast steel divided in halves longitudinally, and fitted with flanges at the horizontal joints to facilitate bolting together for machining. The cylinder is provided with a tongue at the forward end which is supported in a corresponding groove in the end of the turbine casing, permitting the cylinder to expand longitudinally at that end independent of the turbine casing. The cylinder is flanged on the after end and secured to the casing with forty 1½-inch collar studs and nuts. Details of piston and cylinder are shown in Figure 2.

On account of the corroded condition of the groove at the forward end of the casing it was apparent from a previous inspection that it would be necessary to machine the groove and cylinder seating concentric with the center line of the rotor in its normal position in the bearings and also to machine the face of the rotor



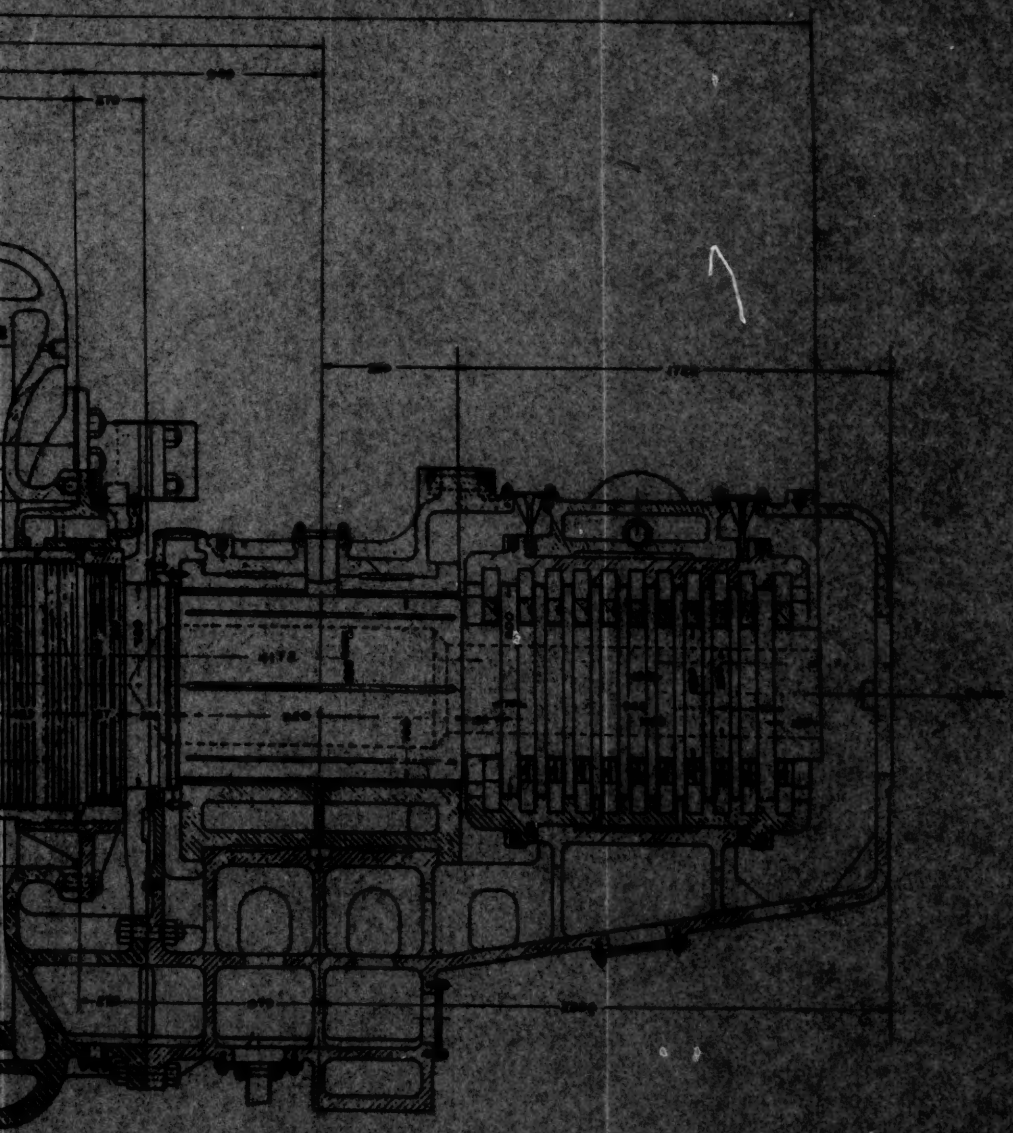


Fig. 1

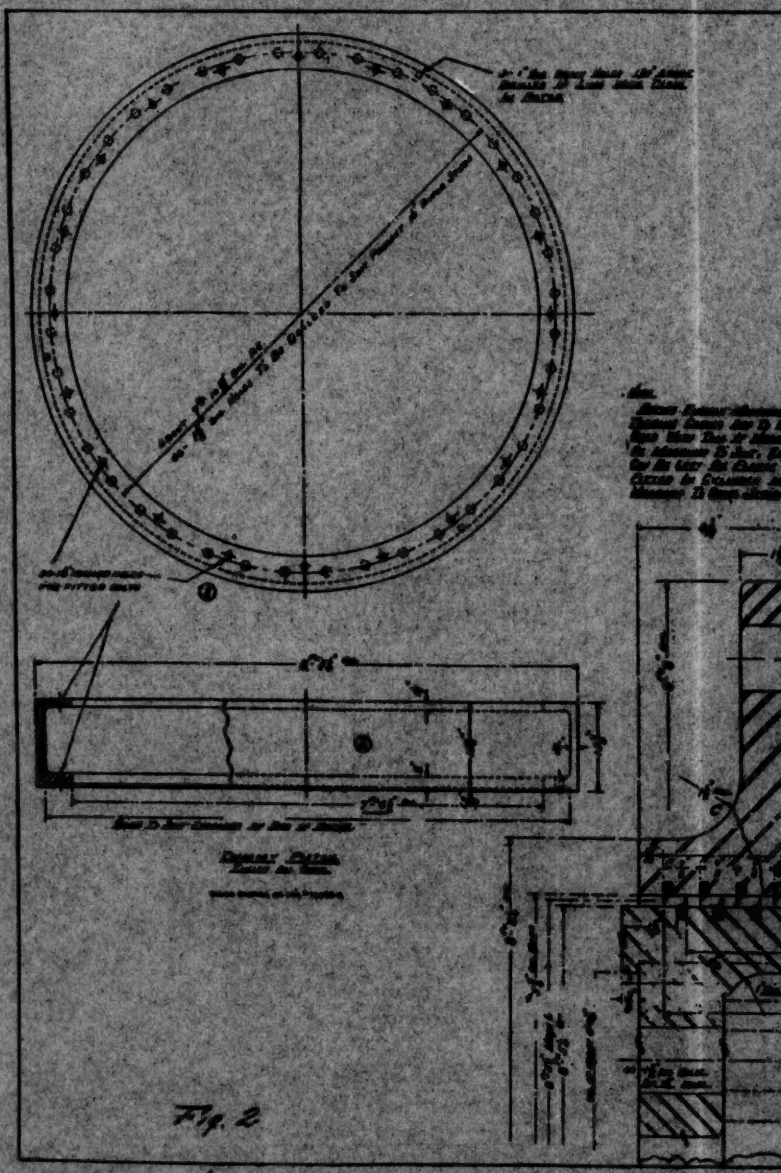


Fig. 2

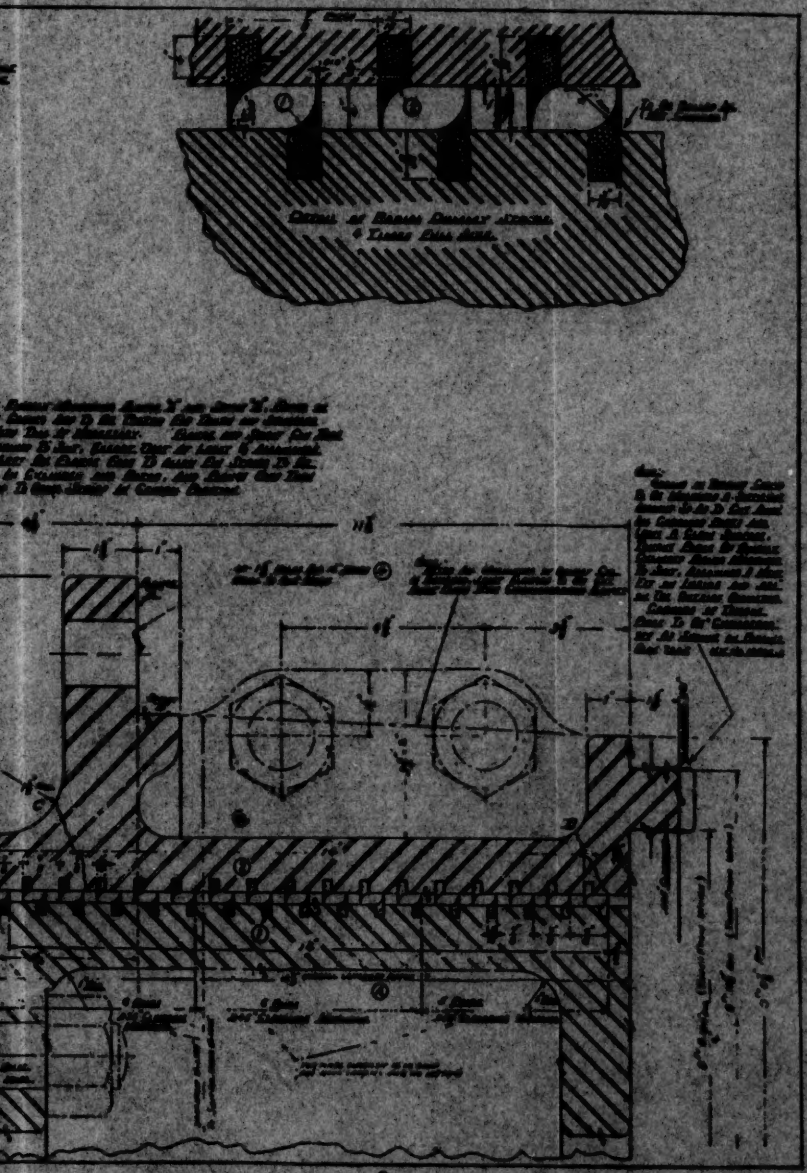


Fig. 3.

wheel and spigot to which the dummy piston is secured in order that cylinder and piston may be properly aligned when secured in position.

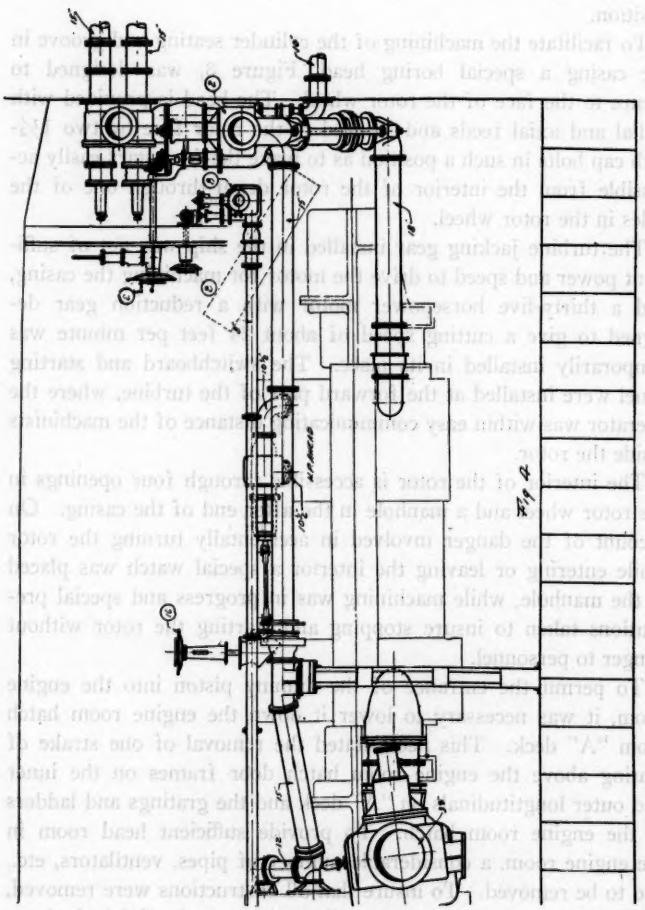
To facilitate the machining of the cylinder seating and groove in the casing a special boring head, Figure 3, was designed to secure to the face of the rotor wheel. The head is provided with radial and axial feeds and secured to the rotor face by two 1½-inch cap bolts in such a position as to make the feed stars easily accessible from the interior of the rotor drum through one of the holes in the rotor wheel.

The turbine jacking gear installed in the ship was not of sufficient power and speed to drive the motor for machining the casing, and a thirty-five horsepower motor with a reduction gear designed to give a cutting speed of about 24 feet per minute was temporarily installed in its place. The switchboard and starting panel were installed at the forward part of the turbine, where the operator was within easy communicating distance of the machinists inside the rotor.

The interior of the rotor is accessible through four openings in the rotor wheel and a manhole in the after end of the casing. On account of the danger involved in accidentally turning the rotor while entering or leaving the interior a special watch was placed at the manhole, while machining was in progress and special precautions taken to insure stopping and starting the rotor without danger to personnel.

To permit the entrance of the dummy piston into the engine room, it was necessary to lower it down the engine room hatch from "A" deck. This necessitated the removal of one strake of plating above the engine room hatch door frames on the inner and outer longtitudinals on "A" deck and the gratings and ladders in the engine room hatch. To provide sufficient head room in the engine room, a considerable number of pipes, ventilators, etc., had to be removed. To insure that all obstructions were removed, a wooden templet of the same dimensions as the finished piston was lowered into the engine room and slipped over the end of the turbine thrust shaft.

On account of the limited head room over the turbine thrust some doubt existed as to the feasibility of installing a solid piston



without the removal of a large portion of the main steam piping, throttle valves, and bulkhead over the turbine thrust. The use of the templet demonstrated that such removals would be unnecessary although there was little room to spare. It will be seen from Figure 4 that in order to pass the piston over the end of the thrust shaft, it was necessary to lay the piston in a nearly horizontal position over the thrust and haul the piston aft as the rotor was raised.

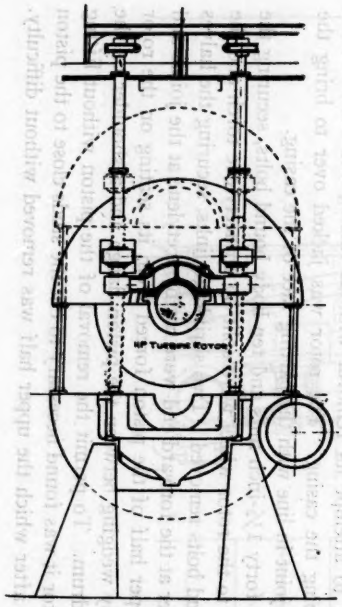
On arrival of the *Leviathan* at Hoboken on December 22, 1926, the work was started and the casing raised sufficiently to check up on the fore and aft dimension of the dummy piston and cylinder. Work on the new parts had been in progress in the shop for some time, but on account of the absence of reliable measurements of the relative positions of the grooves in the dummy piston and cylinder the work of machining the grooves was delayed until measurements could be taken from the work.

The dummy piston and cylinder were then finished to the designed diameters, the packing rolled in the grooves and finished, turned, and bored to a clearance of .035-inch on the first six rows from the steam belt; .045-inch on the second six rows and .055-inch on the last five rows. The final machining was done after the machine work was completed on the ship.

The turbine casing was lifted to a height of six feet, Figure 5(b), placed on stanchions provided for the purpose, and wedged athwartships to prevent surging.

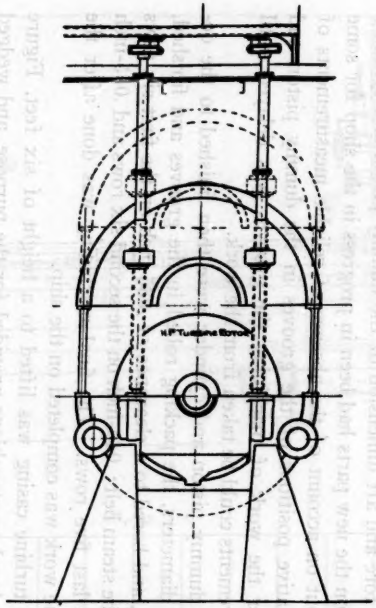
As the old piston was installed in halves it was considered advisable to attempt its removal without raising the rotor before machining the casing. The rotor was jacked over to bring the piston joint in line with the parting face of the casing.

The forty 1½-inch nuts and ten body bound bolts, securing the piston to the rotor wheel were split by an acetylene torch and the nuts and bolts removed. Two shrinkage links, securing the halves together at the forward end were cut by acetylene at the joints and the upper half of the piston forced from its seating on the rotor shell by wedging between the piston flange and a shoulder on the rotor drum. To permit the removal of the piston without lifting the rotor it was found necessary to cut the studs close to the piston flange after which the upper half was removed without difficulty.



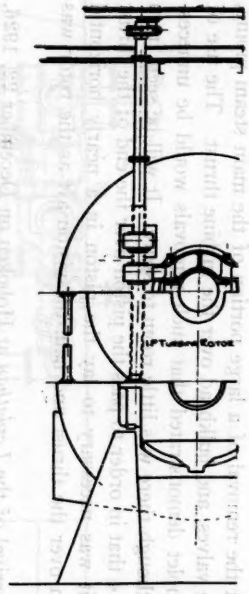
View At After End Of Turbine Looking For'd

Fig 5.a



View At For'd End Of Turbine Looking Aft.

Fig 5.b



View At After End Of Turbine

HIGH PRESSURE TURBINE (AHEAD) ON PORT INBOARD SHAFT.

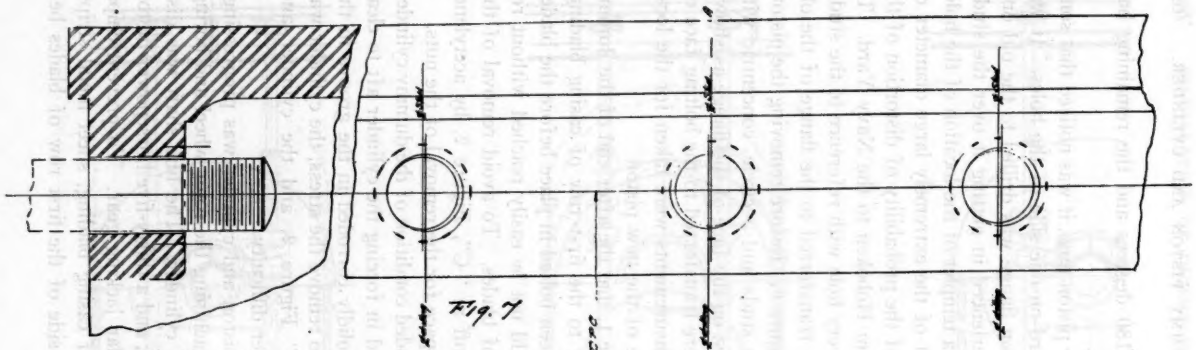
The rotor was jacked over 180 degrees and the remaining half removed in a similar manner.

After removing the dummy piston nuts, it was noticed that some of the studs were binding hard on one side of the holes. It was apparent that if the new piston flange was drilled to the old one, some difficulty might be experienced in getting it over the studs. The usual method of making a templet of the location of the holes was unsatisfactory on account of the extremely large diameter of the pitch circle, and because of the probability of distortion of the templet in transporting it from Hoboken to the Navy Yard. To insure the exact location of every hole with reference to the studs, the location of each stud was transferred to the flange of the old piston by the use of a tool, Figure 6. Before removing the piston, the tool was slipped over the studs and a circle, concentric with center line of the studs, marked on the face of the flange as shown in Figure 7. These circles were transferred to the bolting face of the flange from which the measurements were taken for the location of the holes in the flange of the new piston.

It will be noted from Figure 1, that the after end of the dummy cylinder is in close proximity to the first row of casing blading. Apparently the cylinder had been bolted in place before the blades were installed as the nuts could not be easily reached without removal of the first two rows of blades. To avoid removal of the blades, the cylinder was cut off at "C", Figure 2, by acetylene, permitting sufficient working space for the removal of the nuts.

Due to the warped and corroded condition of the dummy cylinder some difficulty was experienced in forcing the cylinder aft to clear the studs. The tongue was solidly corroded in the groove in the forward end of the casing. To remove the stress, the cylinder was cut circumferentially at "D" Figure 2, and the cylinder and tongue removed without further difficulty.

While the removal of the piston and cylinder was progressing, preparations were made for machining the rotor wheel and boring the casing to receive the new cylinder. The line shaft was disconnected in the dynamo room, and a thirty-five horsepower motor installed in place of this regular jacking gear. To prevent chips from entering the first row of casing blading, sheet metal strips were secured to the forward side of the first row of blades be-



Section through AA

Fig. 7

Fig 6

tween the rotor drums and casing. A canvas covering was triced up over the rotor to prevent foreign matter falling in the blades while machining the rotor wheel.

The studs were extremely tight in the rotor wheel and casing, and could not be backed out by stud removers or wrenches. It was therefore, decided to break them off with a sledge and remove the roots by acetylene after completion of the machine work in order not to delay the finishing of the piston and cylinder in the machine shop.

A compound lathe rest, bolted to a 1½-inch steel plate secured to the horizontal parting face on the lower half of the casing was used for machining the rotor wheel. With the rotor turning at a speed of about one revolution per minute, the rotor wheel face was trued up and the spigot turned concentric with the axis of the turbine. Measurements were then taken for the final machining of the piston. During the machining process, the ships lubrication system was used to lubricate the journals and thrust bearings.

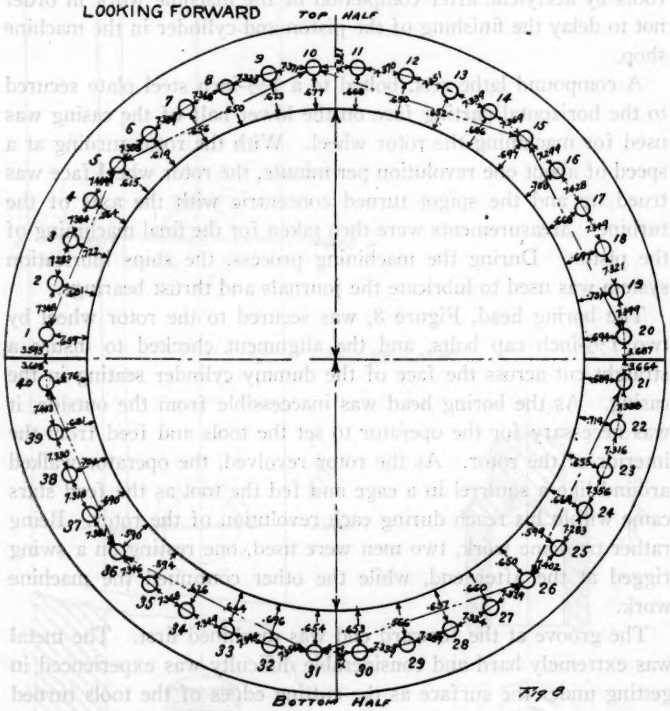
The boring head, Figure 3, was secured to the rotor wheel by two 1½-inch cap bolts, and the alignment checked to insure a straight cut across the face of the dummy cylinder seating in the casing. As the boring head was inaccessible from the outside, it was necessary for the operator to set the tools and feed from the interior of the rotor. As the rotor revolved, the operator walked around like a squirrel in a cage and fed the tool as the feed stars came within his reach during each revolution of the rotor. Being rather tiresome work, two men were used, one resting on a swing rigged at the after end, while the other continued the machine work.

The groove at the forward end was machined first. The metal was extremely hard and considerable difficulty was experienced in getting under the surface as the cutting edges of the tools turned over after a few revolutions. By frequently changing the tools the hard surface was finally broken and the groove roughed out. The face of the casing and cylinder seating were trued up and the groove finish machined.

The top half of the casing was raised and placed on the six foot stanchions. Measurements were taken for the final machining of the dummy cylinder. The thrust bearing and journal caps were

removed and rotor disconnected at the coupling between the high pressure ahead and astern turbines. The rotor lifting gear was connected up and the rotor lifted as shown in Figure 5(a).

The broken studs were then removed. A hole about ¾-inch square was cut through to the bottom of the stud roots by acetylene and the stud quenched with oil. Quenching with oil contracted the



studs and lubricated the threads, permitting the roots to back out more freely. Square taper drifts were driven in the holes and the stud roots backed out with a five-foot wrench. As the acetylene welder had a working space of only 6½-inches for his torch, it is quite remarkable that none of the threads were damaged by cutting process.

The old dummy cylinder was in such a warped and corroded condition that reliable measurements for transferring the location of the holes to the new cylinder flange could not be obtained. As in the case of the dummy piston, the use of a templet was impracticable. It was therefore necessary to take measurements between the studs, and from the cylinder bore to each stud as shown in Figure 8. As a check, pin gauge measurements were taken between every five or six studs. The location of the holes were then laid out on the new piston flange and the distances checked with the pin gauges. When the cylinders were installed the holes were found to be exactly in line with the studs.

The diameter of the piston is approximately eight feet, six inches, which necessitated the raising of the casing to a height of 9 feet to permit the piston to pass inside the casing to its position on the rotor wheel. As the lifting gear is designed to raise the casing to a height of only six feet, extension sleeves, three feet long were fitted to the casing lifting lugs which permitted the casing to be raised slightly over nine feet high.

The casing was placed on six 9-foot stanchions and secured by 3-inch cap bolts screwed in each end of the stanchions through bolt holes in the flanges of the casing. The casing was braced athwartships and fore and aft to prevent surging.

The piston was lowered down the engine room hatch and slung in a nearly horizontal position over the turbine thrust shaft. As the rotor was raised, the piston was passed over the end of the shaft, and gradually worked aft. When the piston cleared the vertical bulkhead over the thrust, the rotor was raised to a height of 4½ feet, midway between the upper and lower halves of the casing. To permit the passage of the piston inside the turbine casing it was necessary to remove the rotor lifting strongback between the lifting screws, Figure 5(b) and to support the rotor forward of the dummy piston, which, at this time was slung in a position just forward of the lifting screws. A saddle, Figure 9, made up of 15-inch × 15-inch oak blocks was placed under the forward end of the journal and the rotor lowered to this position. The strongback was disconnected and removed through the inside of the dummy piston. On account of the weight of the rotor (57 tons) precautions were taken to prevent shifting athwartships in case of accidental listing of the ship.

The dummy piston was then worked in between the lifting screws to the inside of the casing, and secured in position on the rotor wheel with forty 1½-inch nuts. After checking the dummy piston for eccentricity, the holes for the twenty body bolts were drilled and reamed and the bolts driven through the rotor wheel and dummy piston flange from the inside of the rotor. To prevent the nuts from backing off, a 3/16-inch copper wire was passed through holes in the studs and the two ends brazed together. The ends of the body bound bolts were cupped and riveted over nuts.

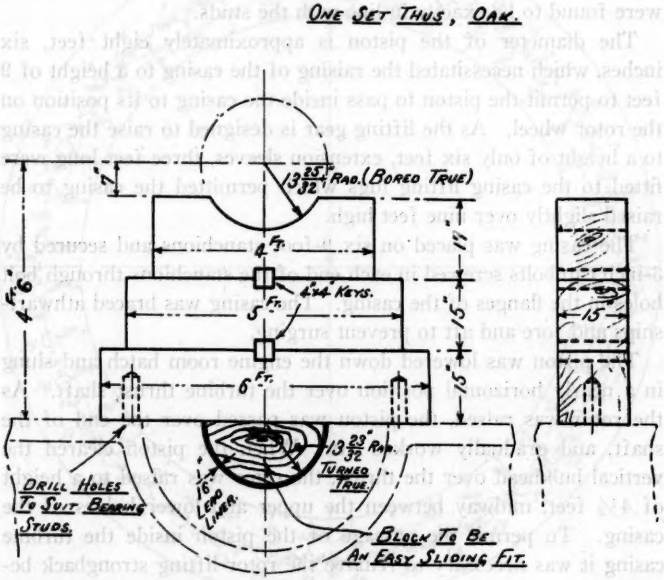


FIG. 9.

On account of the limited working space between the end of the dummy cylinder and the casing blading, it was at first considered necessary to remove the first row of blades to permit the cylinder to be moved aft far enough to clear the casing studs. To avoid the removal of the blading the distance between the bolting face of the flange and the end of the cylinder was made slightly

shorter than the old cylinder allowing a clearance of about ⅛ inch, and permitting the installation of the cylinder without disturbing the blades. The bottom half of the cylinder was lowered into the casing and bolted in position by special socket wrenches made for the purpose.

The rotor was then lowered in the bearings and preparations made to install the top half of the dummy cylinder. A twelve-inch channel iron, fitted at the ends with cross bars about six feet long was suspended under the upper half of the casing by four chain falls outside the casing, and prevented from swinging by two jiggers on each end leading fore and aft. The cylinder was passed under the casing, secured to the channel iron and hoisted to its position on the studs. The cylinder was then secured in place and the channel iron removed.

Leads were taken at eight points on the circumference of the piston as a check on the fin tip clearances and the high spots scraped to the designed clearances.

The work of reassembling the turbine and engine room fittings was completed on February 11, 1927, and the turbine turned over by steam.

Reports from the ship indicate that the installation of the new piston and cylinder has resulted in a more even distribution of the load on the four shafts and a marked increase in fuel economy.

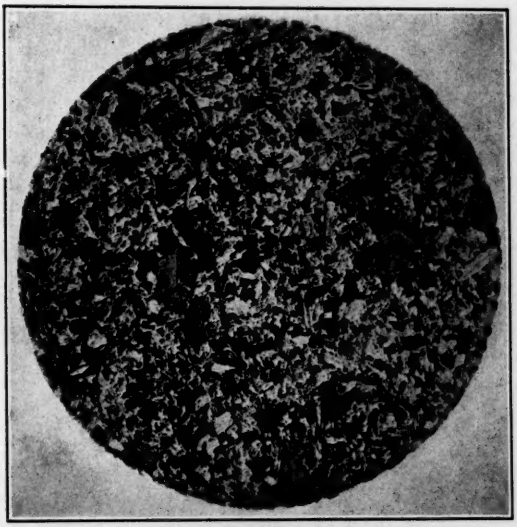
CARBON BRUSHES.

By Lieutenant Commander T. A. Solberg, U. S. Navy,
Member.

The general subject of carbon brushes is one, which to some extent, engages the attention of everyone interested in the design and operation of electrical machinery. The carbon brush is a most important link in the successful operation of all generators and most types of electric motors; and not infrequently, it is the seat of the greatest amount of trouble experienced in the operation of these machines. That its importance has been realized by the brush manufacturer is evidenced by the rapid improvement which has been made in the quality of brush materials during the last twelve or fifteen years. That the designer has also realized its importance is shown by his efforts to design machines which will operate satisfactorily on certain grades of brushes, rather than to design a machine and then depend on the brush manufacturer to furnish or develop a brush which will operate satisfactorily. This latter method is to a large extent responsible for the large number of grades developed by every manufacturer.

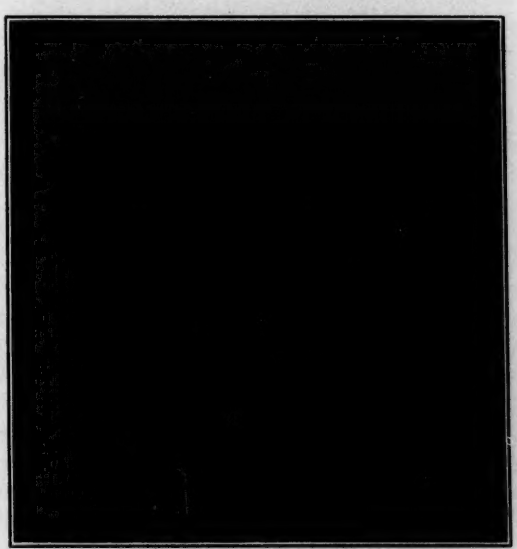
It is the purpose of this paper in general, to cover the physical properties of brush materials and the factors which affect their operation. As a matter of interest, a short history of their development and some of the general methods of manufacture are included. In addition to this it is planned to discuss in a general way the efforts being made toward standardization of brushes.

It might be well at the start to consider the function of the carbon brush. In the term carbon brush is included all brush materials made from combinations of carbon and natural or artificial graphite, and mixtures of carbon, graphite and various metals. In the case of a slip ring on alternating current machinery, the brush by maintaining contact with the rotating ring forms the link for passing a current between the external circuit and the rotating parts of the machine. In the case of a commuta-



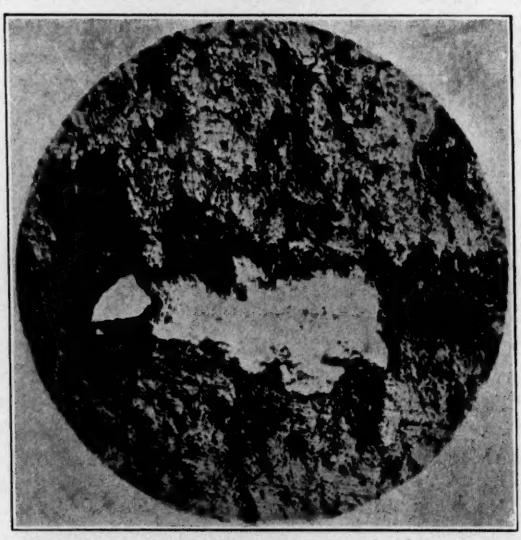
PHOTOMICROGRAPH OF CARBON GRAPHITE BRUSH, MAGNIFICATION 100.

The sharp outlines of four hard inclusions of sand or quartz are very apparent. The white structure is graphite, and the black amorphous carbon.



NATURAL GRAPHITE BRUSH. MAGNIFICATION 300.

This was a high grade natural graphite brush. The transparent white spot in the center of the photomicrograph represents a thin piece of quartz. This inclusion is so thin (probably worn down in polishing the specimen) that the graphitic structure at the edges underneath it can be seen. The large black spots are holes in the surface, which are practically impossible to prevent in polishing.



ELECTROGRAPHITIC BRUSH AT MAGNIFICATIONS OF 300 AND 100 DIAMETERS.

Each picture represents the same portion of the brush. This specimen had been in use and was giving trouble by scratching and cutting the collector. The surface was only lightly polished to remove any dirt or oxide present. The bright, white, lustrous and sharply outlined body on the left is a hard inclusion. The white area to the right of it is copper which was picked up and piled up in front of the inclusion. These photomicrographs clearly show the cause of brushes picking up copper.

tor the function is the same, but the mechanical difficulties are greater because the surface of the ring is broken by segments of mica, or in the case of undercut commutators by recessed grooves. Throughout this paper slip rings and commutators will be referred to as "collectors" unless it is desired to make a distinction between the two.

The earliest method of collecting currents from collectors employed brushes made of copper leaf or gauze. In the copper leaf type, it was necessary to set the brush at an angle away from the direction of rotation, and it was therefore impossible to operate in both directions without special arrangements for reversing the angle of the brushes. Copper gauze was next used and gave somewhat better results than leaf copper. Solid copper was impracticable because it produced severe sparking. All of these brushes, however, gave a great amount of trouble due to the heavy sparking and consequent burning of the commutators. In direct current machines this was caused by the heavy short circuit currents in the coils undergoing commutation, for since this material had a very low contact drop, the total resistance in the commutating coil circuit was extremely low. Since interpoles or compensated windings were not then in use, the short circuit current was necessarily high.

In 1883, Professor George Forbes first suggested the use of carbon for brushes and obtained a British patent but did not actually use his idea. In 1888, Mr. Charles J. Van Depoele in discussing the problem of railway motors with Professor Elihu Thomson, told of his success with carbon as a brush material on stationary motors. At this time electric railways seemed doomed to failure simply because of the brush problem, but the advent of carbon brushes saved the day; and the development of carbon materials for brushes has perhaps done more toward obtaining the successful and efficient operation of electrical machinery than any other factor. About 1895 the idea of using graphitized carbon was developed in Europe but it was not until 1912 that this process was used in the United States. Since that time it appears that European manufacturers have devoted themselves more to the development of the natural graphite and carbon-graphite forms while the American manufacturer has made great strides in perfecting the electro-graphitic or graphitized carbon material. This

material is a graphitic form of carbon obtained by heating amorphous carbon in an electric furnace. Coke, lampblack, tar and various binder materials can be graphitized.

The materials now used in the manufacture of carbon brushes are coke, natural graphite, lampblack, powdered or finely divided copper, and various binder materials, which generally contain tar or pitch. The coke used is obtained by heating the residue obtained from the refining kettles of the oil refineries. It may be from either a paraffin or an asphalt base oil. Natural graphite, which is the crystalline form of carbon, is mined chiefly in Ceylon. The brushes made from these materials or combinations of materials can be divided into classes.

CLASSES OF CARBON BRUSHES.

1. *The Lubricated or Impregnated Carbon.* This class of brush contains coke and small amount of graphite and a binder with some lubricant added to give the desired lubricating qualities. The addition of a lubricant to the brush to improve its characteristics was discovered quite by accident when a dynamo attendant, having dropped a brush in some oil, dried off the oil and then found that it worked better than it did before. At present this class is used very little, for the presence of a lubricant is in most cases undesirable. If high temperatures are reached at any time, the lubricant may liquify and smear the collector. The lubricant will also be used up near the collector surface first and the operation of the brush changes radically with the length of time it is in operation. They may, however, be re-treated when they become dry.

2. *Carbon Graphite.* This contains fine carbon of 200 mesh or over in the form of a good grade of coke with a considerable quantity of either natural or artificial graphite. No lubricant is necessary as the graphite has inherent lubricating qualities. It is found on many modern machines, particularly on reciprocating engine-driven generators and on flush mica commutators. This material is mechanically strong and its characteristics (such as resistance, contact drop, etc.) can be varied by varying the proportions of the different ingredients.

3. *Natural Graphite.* Natural graphite brushes are made of natural graphite held together by a binder which principally is

sugar or molasses. This is used for a binder, because in the baking or kilning process, relatively low temperatures are used which would make pitch undesirable. This material makes a very light brush which is desirable in that it has low inertia forces. For this reason, and perhaps because of its relative softness, it seems to hug the collector better and has less tendency to chatter and vibrate. It is relatively soft, the scleroscope hardness rarely exceeding 20; it generally has some abrasive and cleaning action because of impurities in the graphite; it has a low contact drop, high current-carrying capacity and a low coefficient of friction. Because of the last three characteristics it is much used on turbo-generators and other high-speed machines. The cross resistance of these brushes is generally greater than that measured along the length of the brush because, in the pressing process the crystals form in layers in the long direction of the brush, thereby giving an effect similar to that of a carbon pile. These brushes are mechanically weak and wear relatively fast, the wear apparently being affected by the polarity.

4. *Electrographitic.* This brush material is made of the coke obtained from the residue of Pennsylvania (paraffin base) or Western (asphalt base) oils and a binder. In some grades of this brush, substances are added to give it an abrasive or cleaning action. The resulting mixture is converted into graphitic carbon by heating it to a high temperature in a special electric furnace.

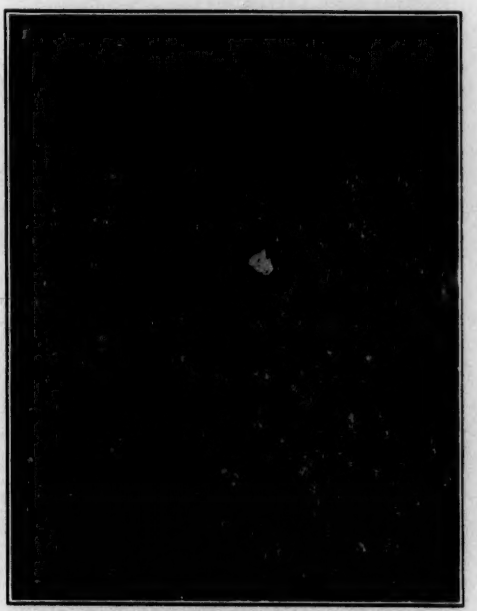
5. *Metal Graphite.* This class of brush is made of powdered copper (200 mesh or finer), graphite, a binder, and in some instances small quantities of other metals, principally lead. Some manufacturers instead of using powdered copper, use a chemical which in the kilning process is reduced to copper, thereby supplying the copper in a very finely divided state. The percentage of metal used varies from about eighty per cent to forty per cent by weight. These brushes find their principal use on collectors of alternating current machinery and on the commutators of low voltage high current direct current machinery.

PROCESS OF MANUFACTURE.

The manufacture of the different classes is very similar and the process will therefore be described in a general way. The coke or carbon used is first calcined by heating until all volatiles have been

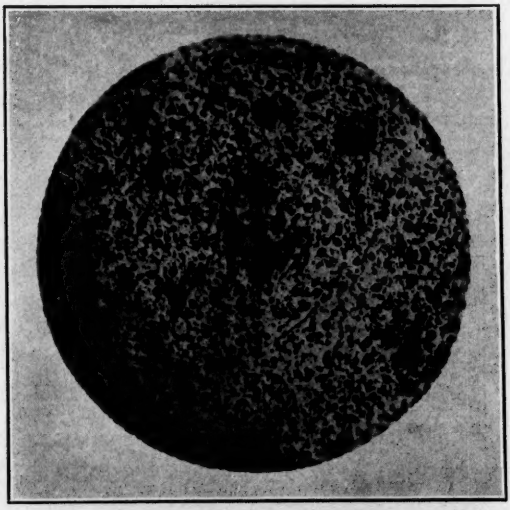
driven off. This process is carefully done because the resulting density and specific resistance is affected by this heating. Numerous samples of this calcined material are tested for these two properties to insure uniformity of product. The next step consists of grinding, which is followed by mixing with the binder and the other materials which are to be added. In some cases the binder and other ingredients are added during the grinding to insure thorough mixing. After mixing, the product is heated just enough to drive off any volatiles present in the binder and the resulting hard dried substance is again reground to about 200 mesh. This fine powder is then compressed into plates or individual brushes in die molds. In the pressing process, pressures as high as 30,000 pounds per square inch are used. The green brushes or plates are then packed in the kilning ovens and carefully covered with sand or fine powdered coke. The heating of the green product covers a period of several days at temperatures between 900 and 1500 degrees Fahrenheit. The temperature is raised gradually, about five degrees per hour, so as to insure that all volatile matter will pass off slowly and thereby prevent any cracking or straining in any of the material. The resulting product is a carbon or carbon-graphite brush material which may, if desired, be graphitized. This is done in special electric furnaces which heat the material to temperatures between 2500 and 3000 degrees Fahrenheit. This temperature, in addition to changing the carbon to the graphitic form, volatilizes and drives off many of the impurities which are undesirable in the finished product. In the manufacture of natural graphite, the pure natural graphite with a binder of sugar or molasses is used and therefore the baking temperature is much lower. Generally, temperatures of from 400 to 800 degrees Fahrenheit are used, this being sufficient to drive off all the volatiles in the binder.

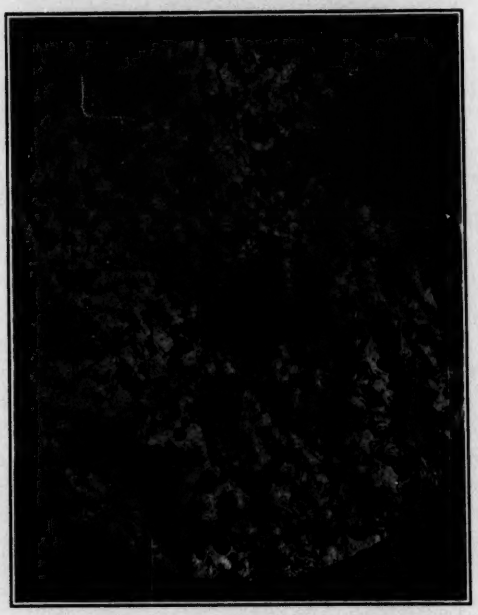
One of the earliest methods of making brushes consisted of heating the final mixture until it was plastic and then forcing or extruding it under high pressure through a die. Cross sections of extruded brushes on examination often show curved surfaces, cleavage planes, and other flaws which are probably caused by uneven expansion throughout the mass as it leaves the die, or by unequal rates of flow throughout the cross section of the material while being extruded. For this reason very few brushes are now made by this method.



A Metal Graphite Brush, Magnification 100.

This brush has a relatively low copper content. The white represents copper, the black carbon, and the bright white spot is an inclusion of copper oxide. There is also a large inclusion about one-half inch from the center of left of picture.





A Metal Graphite Brush at Magnifications of 100 and 300 Diameters.

This sample has a much higher copper content. The picture at 300 diameters, in addition to showing a large inclusion in the lower left corner, is interesting because it shows the characteristic "Newton Rings." This indicates the presence of oil, which was probably added to get a brush with better lubricating qualities.

GENERAL PHYSICAL PROPERTIES WHICH AFFECT BRUSH OPERATION.

1. *Current Carrying Capacity.* The current carrying capacity, expressed in amperes per square inch of cross section perpendicular to the axis of current flow, is the load current which a brush can carry without destructive heating. For carbon brushes this value is from 30 to 70 amperes per square inch, while the metal graphites may have values up to 150 amperes per square inch. This property varies inversely with the specific resistance, which determines the load current heat loss in the brush itself. A low specific resistance indicates high current carrying capacity. This property is also dependent on the following factors:

a. The coefficient of friction, peripheral speed and spring pressure. These three factors are inter-related in producing the total heat of friction between the brush and collector. This heat is developed at the brush face, raises the brush temperature and must be dissipated as well as the heating effect of the load current.

b. Contact Drop: This also produces a heating effect at the brush face and raises the brush temperature. This will be discussed later as a general property.

c. Short Circuit Current: The short circuit current in the coils undergoing commutation produces a small heat loss in the brush itself, and in addition, the loss at the brush face due to its contact drop from collector to brush and back from brush to collector. In alternating current slip ring machines this effect is absent and this may increase somewhat the current carrying capacity of the brush when operated under these conditions.

d. Radiation Factor: This is the ability of the brush to dissipate its heat and it is dependent on its thermal conductivity and mechanical conditions, such as ventilation, brush holder design, etc. There are four paths by which the total heat generated can be dissipated; first, through the commutator, which carries off some of the heat produced by friction and contact drop; second, through the body of the brush, to the pigtail and to the air surrounding top of brush and pigtail; third, from the surface of the brush not covered by the brush holder, to the air; and fourth, through the body of the brush to the brush holder and rigging to the air. It is advocated by many brush holder manufacturers that the holder should come within one-eighth inch of the commutator so as to assist in carrying away heat through this path. But since the

heat transfer from the brush to the holder and thence to the air is necessarily low because of the dead air clearance space between the two sides of brush and holder, it is believed that such holders do not assist in keeping down brush temperatures. They do however accomplish the important object of preventing chattering. It will be generally found that in holders which have part of their sides cut away so as to allow radiation direct to the air, the brushes run cooler than those which are practically enclosed by the metal of the brush holders.

2. *Contact Drop.* The drop in voltage between the collector and brush is caused by the resistance to the flow of current at the point of contact between the brush and the collector. The drop in going from brush to collector is usually slightly less than when the current goes from collector to brush. This characteristic produces a heat effect which varies directly as the contact drop, and hence the desirability of using a brush with low value of contact drop is apparent. The contact resistance, however, also exerts a great influence on the short-circuit current in the coils undergoing commutation. The resistance in this current is made up of the low resistance of the coil, the low cross resistance of the brush face, and the two relatively high contact resistances going to and from the brush. This factor, therefore, has a marked effect on the commutation of any machine and cases of bad commutation may often be eliminated by using brushes having a high contact drop. The value of the contact drop seems to have some relation to the specific resistance of the brush, for materials having a high specific resistance will also have a relatively high contact drop. Metal graphites having approximate contact drop values from .2 to .6 volts per brush while the other classes have values from .6 to 2.0 volts. The contact drop is affected by the following factors:

a. Current Density: Most brush materials show a gradual increase in contact drop with increase in current density within the range of the current carrying capacity. For an increase from 20 amperes per square inch to 60 amperes per square inch the increase in drop will rarely exceed 25 per cent. It should be noted that the contact drop is affected by the area in contact with the collector only insomuch as the area affects the current density. Also, since the current density across the contact is not uniform, the brush drop across the face of the brush is not uniform.

b. Spring Pressure: Increasing the spring pressure (pounds pressure per square inch of brush face area) naturally produces a more intimate contact between the two surfaces and hence decreases the contact resistance and the contact drop. For pressures in excess of four pounds per square inch, the decrease becomes very slight, but between one pound and four pounds the decrease may be as high as thirty per cent.

c. Temperature: Within the ranges of ordinary operation, temperature seems to have no effect on the contact drop.

d. Character of Surface: The contact drop is greatly increased by eccentric and uneven surfaces. Oscillographs of contact drop show that a single flat spot greatly increases the average value of the contact drop. The material of the collector also affects the drop, materials of high specific resistance giving a higher contact drop.

3. *Coefficient of Friction.* This coefficient is the ratio between the frictional force tending to rotate the brushes in the direction of the armature and the pressure normal to the surface of the collector. Its value ranges from .05 to .40, the natural graphite and electro-graphitic grades having the lowest values while the carbon-graphite series have the highest. This characteristic is affected by the following factors:

a. Brush Pressure: An increase in brush pressure increases the coefficient of friction slightly because it produces a better contact. This effect of brush pressure on the coefficient of friction and on the contact drop indicates clearly the importance of obtaining the same brush pressure on all of the brushes of a machine.

b. Peripheral Speed: An increase in speed may decrease slightly the value of the coefficient because of the less intimate contact made with the collector at high speeds. The greatest effect is shown on the hardest brushes because of their greater tendency to jump away from the collector with any slight irregularity of the surface or because of eccentricity and vibration.

c. Temperature: This has no effect unless in extreme cases of local heating the character of the brush surface is changed by the excessive temperature.

d. Current Density: This has an irregular effect on the coefficient of friction. The change in the sound of the brushes when

the load increases is probably caused by the effect of the current on the intimacy of contact between the brush and its collector.

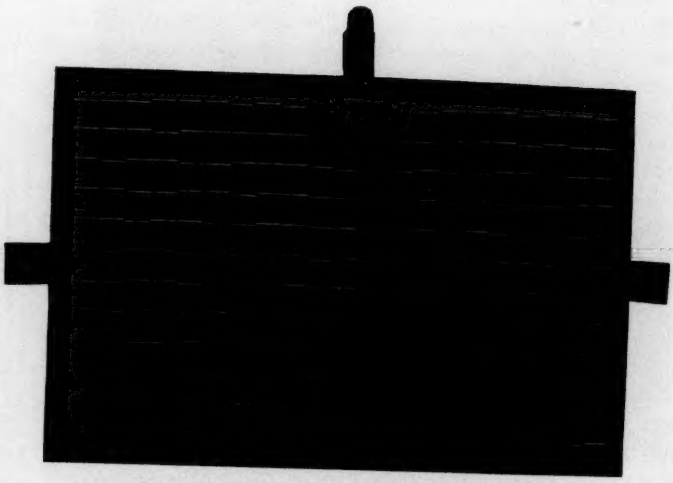
4. *Hardness.* This is a term which is open to discussion. It is the measure of the hardness of the brush material as obtained by the rebound of a diamond pointed weight dropped through a glass tube from a height of about 6 inches in an instrument known as the Shore Scleroscope. The rebound is measured in units on a scale from 0 to 100. This machine is therefore only capable of expressing the *relative hardness of similar materials.* The natural graphites range in hardness from 10 to 20, while the electro-graphitic grades run as high as 70, and in some cases 80. The metal graphites containing a different material run from 5 to 20, which does not indicate that they are softer than the carbon or graphite grades. Hardness does not in any way indicate the strength of the material and its wearing qualities.

5. *Strength.* This property is usually expressed in pounds per square inch as obtained by a transverse breaking test. The test piece is supported at its ends on two knife-edges two inches apart and a force applied by a third knife-edge midway between supports until rupture occurs. The transverse strength is then computed from the formula $\frac{3LP}{2WT}$ where P = pressure in pounds, L = length in inches between supports, W = width, and T = thickness, both in inches. It has been found that if the specimen has its thickness greater than its width, the values obtained are low due to the crushing effect of the knife-edges. A specimen 2½ inches by ½ inch square as used in the Navy tests gives uniform results.

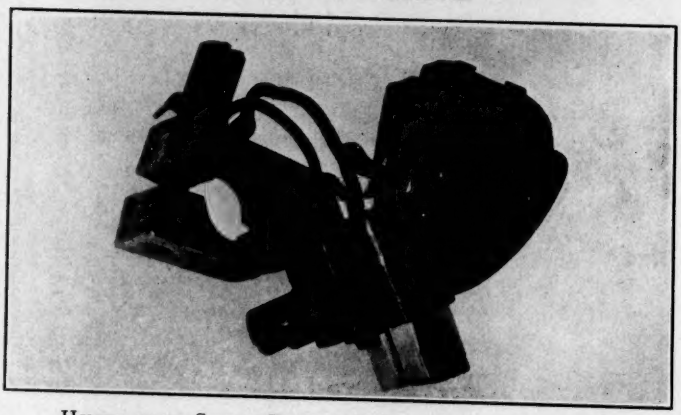
6. *Abrasiveness.* There is no standard method of measuring or determining the degree of abrasiveness except by observation of actual performance. Abrasive material is sometimes added to give better performance on machines with flush mica in order to prevent high mica. Practically all brushes contain some abrasive materials in the form of impurities such as sand, quartz and metallic oxides. The photomicrographs submitted with this paper show some of these impurities. It was noted in examining several grades of brushes manufactured by various companies that even the so-called non-abrasive grades contained some of these abrasive materials. This fact may also be determined by burning the

A METAL GRAPHITE BRUSH AT MAGNIFICATIONS OF 100 AND 300 DIAMETERS.

This sample has a copper content about midway between that of the other two samples. Inclusions of sand or quartz are shown in each. In the 300 magnification the inclusion is a large triangular shape embedded in the graphite.



Box with Spare Brushes.



Holder with Spring Tension Device Down on Brushes.

samples in a suitable furnace and noting the amount and character of the ash remaining. The most abrasive materials contain usually up to 12 per cent of ash, which is siliceous in character. In many of the electro-graphitic grades the percentage of ash runs as low as one-half of one per cent.

7. *Specific Density.* The apparent density, which takes account of the porosity of the material, is usually the one considered. It indicates the compactness of the material when compared to the real density of the substances of which it is composed. The real density of amorphous carbon is about 1.9 and of natural graphite about 2.25. When made into brushes the apparent density becomes about 1.5 for the carbon and 1.95 for graphite. The porosity ranges from 8 to 20 per cent. Too high a porosity may be undesirable under some conditions. The brush may absorb moisture while idle and the vapor formed on heating, incident to severe operating conditions, may cause cracking and chipping. Cases have been known in railway service where brushes have broken into many pieces from this effect. Porosity also lowers the thermal conductivity of the material.

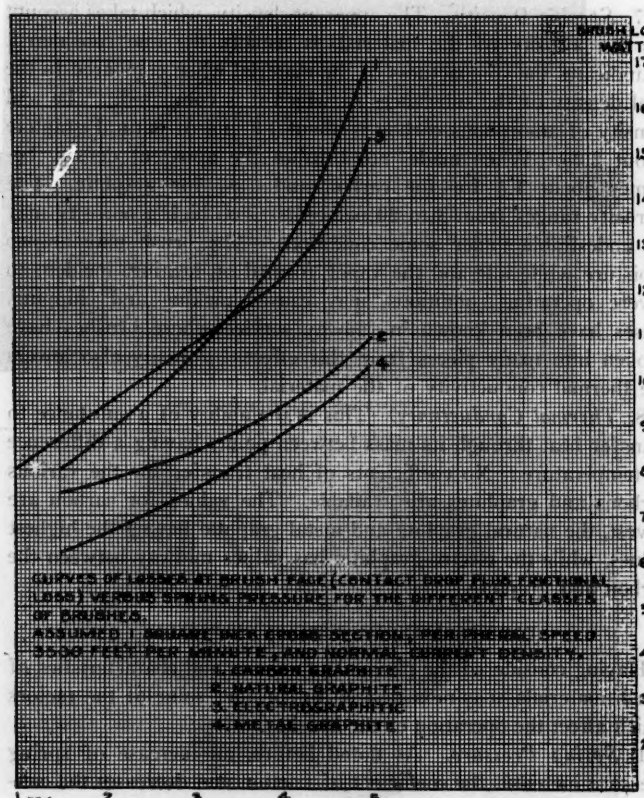
8. *Thermal Conductivity.* This is the ability of a substance to transfer heat through its mass between points having a difference in temperature. No coefficients of heat transfer for the various carbons and graphites are available. The values for carbon are lowest, the natural graphite and metal graphite having the highest values, while the electro-graphitic has values midway between the carbon and graphite.

FACTORS AFFECTING BRUSH OPERATION.

Having considered the physical properties and characteristics of brush materials, it is next desired to consider some of the factors which affect the operation of the brushes.

1. *Spring Pressure.* As noted before, an increase of spring pressure increases the coefficient of friction and decreases the contact drop (lowers contact resistance). It is therefore of the utmost importance that all the brushes on a machine operate with the same brush pressure and it is also obvious that for every machine there is one brush pressure which will give the best operation as regards good commutation and minimum brush losses. Construction of curves of total brush losses—contact drop loss plus friction loss

show that total losses increase uniformly with increasing spring pressure up to 4 pounds and above that point increase more rapidly. In these curves the contact drop loss of the short circuit current was not considered. It would be of such a character as to



raise the ordinates of the curves and have very little effect on the shape of the curves. These curves show clearly that the minimum spring pressure consistent with good commutation and minimum vibration should always be used. Uniform spring pressure on

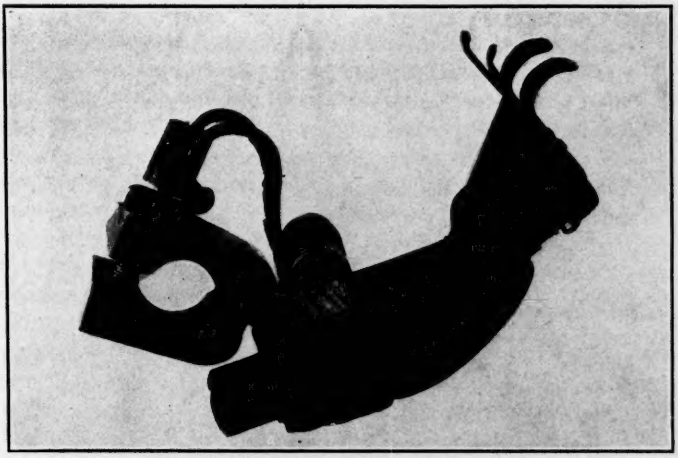
all brushes is of the greatest importance. Unequal pressures will cause the brushes with the greatest pressures (least contact resistance) to carry more than their share of the load current and in cases of overload or other unfavorable conditions of commutation may cause bad commutation. It is common to find the same spring tension setting on all the brushes of a machine. There then exists a difference in pressure between a brush on the top center and bottom center of the ring equal to twice the weight of the brush. In the case of the heavy metal brushes this is considerable.

There exists under certain conditions a partial vacuum beneath the brush and this increases the effective pressure of the brush on the collector. This phenomenon has often been observed in the laboratory on special testing machines where there is a minimum of vibration and brushes are carefully bedded in. It is not believed that this occurs often outside of the laboratory because of vibration and other disturbing influences which reduce the possibility of a vacuum being formed. In this connection it is interesting to note that one large manufacturer of electrical machinery has made extended experiments on slotting the surface of the collectors of alternating current machinery with the idea of preventing the formation of either a vacuum or a pressure underneath the brush. This slotting of the collector resulted not only in more uniform brush pressures but also in a marked decrease in the brush wear.

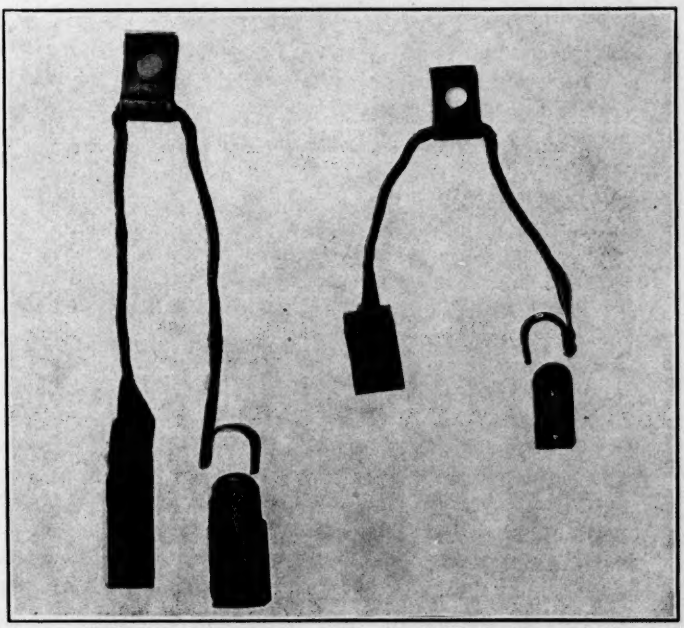
2. *Brush Holders and Brush Angles.* Brush holders are very important in securing good brush performance. The sliding box type of holder is in most general use. It may be constructed as a radial holder with the brush axis held normal to the commutator, or constructed to make an angle with the commutator. This latter type is known as the reaction type because the force due to friction has a component which increases or decreases the effective pressure of the brush against the commutator. For this type of holder the contact surface of the brush is bevelled and the toe of the brush will be considered under all conditions to be the edge at which the acute angle of the bevel is formed. The angle the brush axis makes with the normal to the collector is the angle at which the brush is running. If the collector rotates toward the toe (from toe to heel) the brush is said to be operating *leading*,

and if the opposite, *trailing*. In the former case one component of the friction force opposes the spring pressure while the other component forces the brush against the face of holder. In the trailing case, one component is added to the spring pressure while the other pushes against the holder.

Radial types have the disadvantage that they are more prone to chatter and vibrate, and the brushes frequently tilt and wear at the edges. The reaction holders generally give smoother action with less chattering and screeching. The proper angle at which a brush should operate has always been in dispute. Operation in the leading position is gradually becoming the more favored method, at least for high angular speeds, as this seems to give less vibration and less danger of chipping due to surface irregularities, eccentricity of collector, etc. When operating trailing, the best angle seems to be about 15 degrees as this small angle affects the spring pressure only slightly. For angles greater than 15 degrees the top face of the brush is also bevelled so that one component of the spring pressure will hold the top of the brush against the same side of the brush holder that the bottom of the brush is held by one component of the friction force. When operating leading, the angles are larger depending on the speed. For speeds up to 4,000 feet per minute, angles of 20 degrees are usual, while for greater speeds the maximum angle used is 45 degrees. There is a great diversity in practice and there appears to be no universally best angle adopted by the different manufacturers. In the leading operation it is to be noted that the components of the spring pressure and the friction, which tend to force the brush against the front and back of the holder, respectively, oppose each other. If the brush angle (angle from radial position) equals the angle whose tangent is the value of the coefficient of friction for the brush they will balance each other and hence the brush may chatter. For this reason the running angle should never be less than 20 degrees to prevent any possibility of this occurring. The tangent of 20 degrees is 0.364, which is a high value of the coefficient of friction, the ordinary upper limit being about 0.3000. The brush holder selected should be one which will always hold the brush against the collector at the desired angle without tilting or canting in the holder and thereby



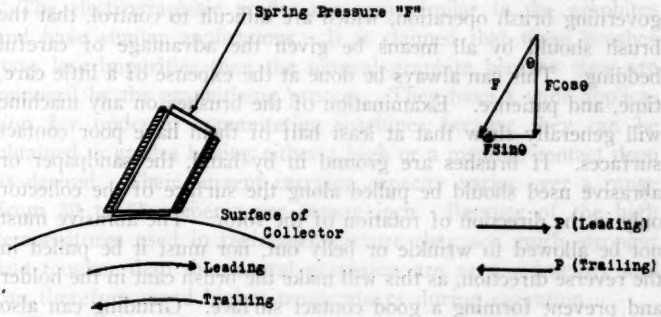
HOLDER WITH SPRINGS RAISED FROM BRUSH.



SHOWING METHOD OF ATTACHING SHUNTS OR PIGTAILS.

giving irregular wear and poor contact surface. The brush holders and the brushes should have enough clearance to give a good easy sliding fit at all temperatures up to 100 degrees centigrade.

3. *Surface of Collector.* Commutators are practically always made of copper. Slip rings, however, are made of various materials and it should be remembered that materials having the higher specific resistance produce a higher contact drop. Some



F = Spring Pressure

FCOSθ = Component of Spring Pressure Normal to Collector

FSINθ = Component of Spring Pressure Tangential to Collector

u = Coefficient of Friction

P = Force of Friction Tending to Rotate Brushes in Direction of Collector

By definition $u = \frac{P}{FCOS\theta}$ or $P = uFCOS\theta$

If u = tan θ, then P = F tan θ Cosθ = F Sin θ

Therefore, in leading operation, P and F Sin θ balance each other and may result in chattering. In the trailing case, P and F Sin θ act in the same direction to force the brush against one face of the collector.

attention should also be given to the heat treatment of slip rings so as to insure a good crystalline homogeneous structure. Photomicrographs of different rings often show coarse-grained, uneven structures which undoubtedly increase the brush wear as well as the contact drop loss and friction losses on the rings. It is need-

less to state that the surface of the collector should be as nearly perfect as possible. A single flat spot on the collector raises the contact drop and in addition may cause sparking with its consequent bad effects. Eccentricity, high or low bars, high mica, roughness, etc., have bad effects on the performance of the brushes as well as on the performance of the machine itself.

4. *Surface of the Brush.* The surface of the brush itself is too often given insufficient attention. There are so many factors governing brush operation, which are difficult to control, that the brush should by all means be given the advantage of careful bedding. This can always be done at the expense of a little care, time, and patience. Examination of the brushes on any machine will generally show that at least half of them have poor contact surfaces. If brushes are ground in by hand, the sandpaper or abrasive used should be pulled along the surface of the collector only in the direction of rotation of the rotor. The abrasive must not be allowed to wrinkle or belly out, nor must it be pulled in the reverse direction, as this will make the brush cant in the holder and prevent forming a good contact surface. Grinding can also be accomplished by fastening a strip of sandpaper completely around the collector and then turning the machine over a few revolutions by power. With this method it is often necessary to do the final surfacing by hand with a fine sandpaper. After being bedded, all dust should be carefully removed and the machine run light or under a light load until the brushes have become well seated and acquire a good polish.

APPLICATION AND SELECTION OF BRUSHES.

With the large variety of grades of brushes carried by the different manufacturers it is a problem to determine what grade to use on a machine. Usually the machine will perform satisfactorily with the brushes originally furnished and in this case it is only a question of renewal when these are worn out.

As a rule, the carbon-graphite grades are used on medium speed commutators, either flush or undercut. In the case of an undercut commutator a grade with minimum abrasiveness should be used. These brushes have a carrying capacity up to 50 amperes per square inch.

Graphite brushes find their greatest application on high speed high current machinery because of their high current carrying capacity (up to 70 amperes per square inch), low coefficient of friction and low inertia force. They are used considerably on slip rings of turbo-generators, synchronous converter commutators and high speed direct current generators. These brushes because of their lightness have low inertia forces and therefore hug the collector very well.

The electrographitic grades are very similar to the graphites and have similar applications. It is claimed that these brushes have less impurities than the natural graphite because they are removed by the graphitizing process. They have a wide application for undercut commutating machines because they can be obtained in grades having either a high or a medium contact drop as desired. Their current carrying capacity varies over a range from 30 to 65 amperes per square inch. Because of the high temperatures used in their manufacture they are much stronger and tougher than the natural graphites, are very refractory and can therefore stand higher temperatures during operation.

The metal graphite brushes are used wherever a low contact drop and a high current carrying capacity is necessary. Their greatest application is on low voltage high current apparatus, and on the slip rings of turbo-generators, synchronous converters and induction motors. These brushes have the disadvantage of producing a fine copper dust as they wear, and this dust is deposited in the windings of the machines; in the case of converters, the dust is also deposited on the commutator, the brushes and the brush rigging, which is a bad feature. In a great many cases also, this class cuts the slip rings considerably. Despite these disadvantages, on many machines this grade must be used because of its low contact drop losses; for, with a carbon or graphite brush this loss would be so great that the rings and brushes could not dissipate the heat evolved and the brushes would fail by destructive heating. Also with a carbon or a graphite brush, at least double the number of brushes would be necessary because of their relatively low current carrying capacity. The metal graphite grades have capacities from 75 to 150 amperes per square inch, depending upon the percentage of metal in the brush.

As a rule, brushes of different grades should not be mixed on any one machine. In some cases, in order to obtain desired lubricating properties, it may be desirable to use one graphite brush on each holder arm, the position of the graphite brush differing on each holder, so that the whole surface of the commutator will be covered. In such cases, the best lubricating qualities will be obtained if no shunts are supplied to these brushes. Otherwise, all the brushes on a machine should be of the same grade. The use of different grades on the same holder arm will naturally result in unequal currents being carried by the different brushes; and using brushes of one grade on one arm of a polarity with a different grade on another arm of that polarity will cause unequal loads on these two arms because of the difference in resistance in the two circuits, caused principally by the difference in contact resistance of the two different grades used. The same effect is obtained when using different spring pressures which also unbalances the resistances in the circuits.

In considering the selection of brushes careful attention should be given to the shunts, since these must carry practically all of the current from the brush to the terminals on the machine. Shunts are made of fine copper wire twisted or braided together so as to form a strong flexible connection. The best material seems to be a triple twisted shunt, which possesses maximum flexibility and mechanical strength, and gives the least trouble from ravelling when any of its strands are broken. The shunt must not only have enough cross section to carry the maximum current but must be large enough so that it will pick up this current at the brush connection. There is a small contact drop at this point and unless the shunt is of sufficient size and the connection well made the shunt will eventually fail. There are numerous methods of connecting the shunt to the brush. The majority of these consist of fastening the shunt to the brush by means of bolts and nuts, screws, or rivets,—with or without solder. These methods have the disadvantage that they add to the weight of the brush and thereby increase its inertia forces, and also that the surface contact at the brush may vary during the life of the brush. Of this type, the best seems to be one in which the bolt is threaded through the brush, thus insuring better contact than a smooth bolt or rivet. A recently developed connection consists of

drilling the brush to the desired depth, inserting the end of the shunt, preferably with the end enlarged by unravelling slightly and turning back the ends of the strands, and then tamping around the shunt a metallic powder or cement. If properly done this shunt gives a very low voltage drop (generally .01 volt), is dependable throughout the life of the brush, is mechanically strong and adds but little to the weight of the brush.

CARE OF BRUSHES.

One of the prime requisites of good brush operation is *cleanliness*. The brushes, brush holders, brush rigging, and collectors should at all times be kept clean. If dirt is allowed to collect, the brush may stick in the holder and result in breaking chips off the brush, as well as damaging the collector. The brushes should occasionally be moved by hand to see that they are free in their holders. A hand bellows is a great assistance in keeping a machine clean. Where air pressure is available the machine should be blown out carefully before starting. Covering the collector end when the unit is not in operation also assists in keeping out dirt. Other points in the care of brushes, such as maintaining a uniform spring pressure, care in grinding, etc., have previously been mentioned.

UNDERCUTTING COMMUTATORS.

This practice, which was first commenced in 1905, has greatly simplified brush problems. It consists of slotting or grooving the mica between the commutator segments to a depth of about 3/64ths of an inch so that the brushes need not have any abrasive qualities to wear down the mica. In the ideal case the mica is of such a composition that it will wear at the same rate as the copper. This was possible with the older machines when a grade of soft yellow mica, sometimes known as amber or India mica, was used. Since this grade of mica has become scarce and expensive, the cheaper and harder grades have been substituted. This invariably gave trouble because of high mica and for this reason undercutting was hit upon as a remedy. On such commutators better contact exists and the general performance of the brush is improved. Among the other advantages are: an increased life of the commutator and brushes; elimination of flat spots, flashovers and

excessive sparking; decrease of heating; and much quieter operation. The only disadvantages are: accumulation of dirt or oil causing shorts between bars, which is easily prevented by a little care and attention; expense and work of reslotting when copper has worn down. There are on the market, however, many machines for reslotting, and in most cases the work can be done at little expense. A short piece of fine toothed hack-saw blade attached to a suitable handle can often be used and will give excellent results. A single edged cutter should never be used as it tends to crack and tear the mica. After undercutting, the edges of the commutator bars should be slightly bevelled to prevent any danger of the sharp edges cutting the brushes. There are two shapes of grooves used—the square and the "V" shape. The first is much easier to cut and is in more general use. The "V" shape can be cut by first using a hack-saw blade or small circular saw to cut just below the edge of the bars and finishing the groove with a small three-cornered file. At the present time, at least one large manufacturer of electrical machinery in this country now builds all commutators with recessed mica, while many others resort to this on all machines above certain sizes.

STANDARDIZATION OF BRUSHES.

This is a subject which presents a great many difficulties, but nevertheless efforts are being made to effect some gradual standardization of brushes. In England an attempt is being made to adopt a brushholder of the reaction type as a standard, and with this can naturally be associated brush standardization. It is interesting to note the steps which had been made in this line in the German Navy as early as the beginning of the War. A standard brush holder of one size only was used throughout their service and in addition only two grades of brushes were used. There are included with this paper photographs of one of these holders which was removed from a ten H. P. motor of a German battleship subsequent to the armistice. This type of holder was found on practically all the motors of different men-of-war, including submarines. The photographs illustrate the following points:

(*a*) A large number of relatively small units. This is of course preferable to large units, since better operation invariably occurs

in the former case because of better total contact surface, less vibration, etc.

(*b*) Each unit could be removed from the brush arm without in any way disturbing the other units. In removing a unit the through bolts are so arranged that they remain on the holder with no danger of dropping off on the armature.

(*c*) Simple rugged construction.

(*d*) Light springs in box to keep brush permanently against one side of box and still allow freedom to slide in box.

(*e*) Simple method of attaching pigtails. Clipped on to the brush. Not a particularly good method as contact drop is rather high.

(*f*) Since there were only two grades of brushes used, the small box shown carrying extra brushes provided ample spares for a large number of motors.

Although there is at present no universal brush for all kinds of service it is believed that most machines could be designed to operate on certain good grades of brushes having certain characteristics. Observation of the electrical machinery installation on some of our recent battleships discloses the fact that many motors of various sizes are using grades of brushes which, although having the trade-marks of different companies, are brushes of very similar characteristics. The writer has also observed that in one of our Navy Yards it has been the practice during the last few years to purchase carbon plates of 2⅞ inches by 4½ inches and various thicknesses and to cut the renewal brushes out of these plates. Pigtails are secured by a washer ring, through bolt and nut. It is also of great interest to know that the same grade of brush material has been used for the past year in making these brushes, and their operation on the various motors in the shops has been thoroughly satisfactory. The grade used was an electro-graphitic material having a wide range of application.

The problem of obtaining renewal brushes in the Navy has always been a lengthy as well as a difficult procedure. As a rule, proprietary requisitions were made on the manufacturer of that particular machine, or requisitions made specifying the identical grade of brush originally supplied with the machine. With the large number of various sized electrical machines installed on a ship, it was necessarily expensive to maintain spare brushes

of all the different sizes and grades used on a ship and manifestly impossible to maintain stocks at Navy Yards or Supply Depots which could supply any machine on any ship. Sister ships for instance are not necessarily equipped with the same electrical apparatus.

In order to devise some method of simplifying the situation as regards the supply of carbon brushes the Bureau of Engineering kept elaborate records of the brushes used on all ships, especially those used on the ship's generating sets and other large machines. By a systematic arrangement of all this data into groups, it was naturally found that a great many machines made by different manufacturers used brushes of similar characteristics. From a careful study of the records made and by actual trials on several ships, it was determined that the needs of the Navy can be satisfied by using only five different grades of brushes, and that three of these grades will satisfy most requirements. The process of discarding the old system and limiting the service to the use of these five grades will of necessity be a gradual one. It will, however, not only simplify the procurement of brushes by individual ships, but will reduce greatly the time previously lost in obtaining new brushes, for the reason that with only five grades to be supplied, it will be possible now to maintain stocks of these in various sizes at the different Supply Depots and Navy Yards. It also will reduce considerably the cost of brushes, as it will be possible to buy in large quantities instead of in small lots on ships' requisitions as heretofore.

From the above instances, it can be seen that in the cases of large organizations using large quantities of carbon brushes, some attempts at standardization have been made. The Electric Power Club has done some work on this problem in this country. In England considerable work has been done towards developing a standard brush holder and with this, brush size standardization naturally can be associated. The standardization of both brush sizes and brush holders is, no doubt, possible, but it will naturally be a slow process. It would result in considerably cheaper brushes as well as brush holders, for each different brush size generally means small lot manufacturing with special dies and special apparatus.

As to a standardization of the grades of brushes in each class, this seems a much more difficult undertaking. The large manufacturers list from 25 to 35 grades of brushes. The reason for this large variety is undoubtedly the existence of relatively small number of freak machines, which, because of poor manufacture or non-uniformity of materials, will not operate with the same brushes as sister machines built to the same specifications; and secondly, by the whims of certain users who insist on a special grade of brush. Data collected by the Navy, also shows that the large manufacturers of electrical machinery use only one or two grades of each class of brush in equipping their output. One large builder of electric motors had, as early as 1914, limited his use of brushes to three grades, two of which were carbon-graphite and the other a natural graphite suitable for very high speeds. It therefore seems possible for the manufacturer to produce machines which in practically every case will operate on the desired grade of brush. It would also seem practicable that machines of the same general size could be equipped with brushes of a standard size, although the machines were built by different manufacturers. In the unusual cases of freak machines, mentioned previously, a special grade is resorted to in order to cover up its deficiencies. For this reason it is not unlikely that at some future date the specifications for electrical machinery will require that the machine operate satisfactorily on certain accepted grades of brushes of some standard size.

SALINITY OR IMPURITY DETERMINATION OF BOILER FEED WATER.

By C. Huey, Civil Member.*

1. Introduction.

The determination and control of the amount of impurities in boiler feed water has, in recent years, become a matter of great concern for those responsible for the maintained efficiency of the steam propelled ships of the Navy. The problem of obtaining suitable feed water is somewhat narrowed by the general instructions which discourage the use of fresh water from ashore, either undistilled or distilled. Water from the condensation of evaporated sea water is, therefore, assumed to be the character of the boiler feed water to be experienced in all Naval ships. On this basis, this article analyses the character of the feed water, and describes the various methods of quantitively measuring the impurities. Also the closely related electrical methods are described for determining the concentration of sea water in evaporator or distilling systems.

2. Feed Water Impurities.

Minor impurities may be found in boiler feed water, such as minute quantities of iron and aluminum salts, silicates, mineral and organic acids, grease, carbon dioxide and air. These impurities are usually derived from lubricating oils and from the products of corrosion in boiler and water stowage spaces. Even if present in considerable quantity, these impurities would not of themselves form boiler scale. The air and the acids promote corrosion, but the former can be eliminated by heating the feed water, and the latter may be neutralized by keeping the water slightly alkaline.

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By far the most troublesome impurities are those derived from sea water which enters the feed water through inefficient or improperly operated evaporators, or through leaks in the condensers, or at other points where sea water adjoins the fresh water lines.

3. SEA WATER IMPURITIES.

The quantities and proportions of the various impurities contained in sea water vary in different parts of the world. In the polar seas the total impurities average about 2.9 per cent by weight of the water, at the equator the average is about 3.55 per cent, in the Mediterranean and other enclosed bodies of water, local conditions produce some variation. It is generally considered, however, that the total impurities in sea water will average 1/32 part by weight of that water.

The character of the impurities also varies to some extent, although they always include sodium chloride (common salt), and the salt of magnesium, calcium and potassium. Data have been collected on the ingredients of sea water in all parts of the world. From the average of these data as a basis, and assuming a total impurity content of 1/32, the table, Plate I, showing the names, proportions and properties of the ingredients usually found, has been worked out by several development engineers.

4. "HARD" WATER.

It will be noted in column 8 of that table, that the term "Hardens Water," is used to describe one of the properties of the magnesium salts. For many years the term "hard," has been used to describe a water which forms deposit or scale on being heated, or which refuses to lather freely with soap. The term "temporary hardness," is used to describe that produced by the carbonates of magnesium and calcium held in solution by dissolved carbon dioxide which evaporates upon boiling and precipitates the insoluble carbonates. The term "permanent hardness," is used to express that caused by the sulphates and chlorides of magnesium and calcium which do not precipitate until a high concentration and high temperature are reached. The degree of hardness is expressed in terms of grains of hardening compound per gallon of water.

5. CAUSES OF SCALE.

An analysis of boiler scale always shows that it is composed principally of compounds of calcium and magnesium. Other substances are always very small in comparison with the quantities of calcium and magnesium compounds. This fact has led to the conclusion that these two elements are the cause of scale. By eliminating these two elements, it has been proven by experimenters, that the water will no longer form scale.

When water containing the compounds of calcium and magnesium is heated to boiler temperature and under boiler pressure, the solubility of these substances is reduced. Some of these substances, notably calcium sulphate, becomes insoluble at about boiling temperature. This causes them to revert to solid form and they precipitate onto the tubes or heating surfaces, as scale or sludge.

Other substances, such as the sulphate and chloride of magnesium, remain in solution longer. When the water evaporates, these compounds remain behind. If new water containing these compounds is fed in, to take the place of that which is evaporated, there will be a gradual accumulation in the boiler until the conditions of temperature, pressure and impurity are such, that no more impurities can be dissolved. When this saturation point is reached, further addition of these impurities will cause precipitation in the form of scale or sludge.

6. EFFECT OF SALT (SODIUM CHLORIDE, NaCl).

It will be noted from the table given in Plate I, that no injurious effect can be directly attributed to the presence of common salt other than that of rendering the water unpalatable for drinking purposes. It is probable that large concentrations of NaCl would contribute to priming, although several authorities state that priming is brought about by the *combination* of high concentrations of soluble impurities, such as sodium salts, with finely divided insoluble matter held in suspension in the water.

In the Zeolite water softening process, now manufactured by numerous concerns and used for feed water treatment by a large number of power plants, the sodium salts, including NaCl, are deliberately substituted in place of magnesium and calcium com-

INGREDIENTS AND PROPERTIES OF AVERAGE SEA WATER.

	1 Form	2 Chemical	3 Common Name	4 Proportion in % of total impurities	5 Grains per gallon in 1/32 Sea Water	6 Grains for 1 grain reading on water impurity meter	7 Solubility in Grains per gallon at 60° F.	8 Properties
HARMLESS	$NaCl$	Sodium Chloride	Salt	79.17	1444	1.60	18,500	These compounds do not destroy soap or form scale. Their effect in causing priming is doubtful. In concentration exceeding 5 grains per gallon they render water unsuitable for drinking.
HARMLESS	K_2SO_4	Potassium Sulphate		2.45	15	.05	5,700	
HARMLESS	$NaBr$	Sodium Bromide		.06	1	.001	46,600	
	TOTAL OF HARMLESS COMPOUNDS			81.68	1490	1.65		
HARMFUL	$MgCl_2$	Magnesium Chloride		9.84	180	.20	27,000	Hardens water. Extremely corrosive.
HARMFUL	$MgSO_4$	Magnesium Sulphate	Epsom Salt	5.12	93	.10	20,000	Hardens water. Corrosive under most conditions.
HARMFUL	$CaSO_4$	Calcium Sulphate	Gypsum	3.36	61	.07	120	Forms hard, slate like scale.
	TOTAL OF HARMFUL COMPOUNDS			18.32	334	.37		
	TOTAL OF ALL IMPURITIES			100.00	1824	2.02		

COMMENTS.

Column 4—Shows the percentage relation of each impurity to the total impurities. It was compiled from data supplied by the U. S. Navy and agrees approximately with the figures obtained by the British Government's Challenger expedition at 362 ocean stations. The Challenger observations showed that the average relative values of the impurities are as follows: NaCl 77 per cent; $MgCl_2$ 10 per cent; $MgSO_4$ 6 per cent; $CaSo_4$ 4 per cent; KCl 2 per cent; balance of 1 per cent in traces of various other salts.

It is usually assumed that the total impurities are 1/32 or 3.125 per cent of the sea water. This relation varies, however, from 2.7 per cent to 3.7 per cent in the open sea and between somewhat wider limits in closed waters such as the Baltic and Mediterranean.

Column 7—NaCl (common salt) is very stable under heat and pressure. When heated it remains unchanged until a temperature of 772 degrees C. is reached. It then goes into vapor but still as NaCl. K_2SO_4 decreases in solubility after passing a temperature of 32.4 degrees C. but never enough to cause precipitation in the concentrations in which it is found in boiler water. $MgCl_2$ is highly corrosive because under the influence of heat it reacts with water to form magnesia (a well known heat insulator) and hydrochloric acid. $MgCl_2 + H_2O = MgO + 2HCl$. The acid exercises a direct decomposing action on the iron of the boiler and an indirect action by promoting electrolysis.

pounds. It is customary where using water softened by this process, to allow the boiler concentrations of the sodium salts to reach 500 to 1000 grains per gallon, before blowing down. This is considered to be practical proof of the relatively harmless character of the sodium salts.

7. EFFECT OF SCALE.

Scale on boiler tubes has two principal effects; first, by acting as a heat insulator between the water and the heated tube, it permits overheating and burning of the tube; second, by acting as a heat insulator between the heated tube and the water, it interferes with the transfer of heat to the water and thus lowers the steaming efficiency of the boiler.

When impure water is used, there are two other sources of loss which increase the expense of operation. These are the losses due to repairs to the boilers, including the mechanical removal of scale, and the loss of heat units caused by the frequent blowing down which becomes necessary to prevent the concentration of impurities in the boiler from rising to dangerous proportions. Far more serious than any of these money losses or delays for repairs and cleaning, is the loss of military effectiveness which would occur on Naval ships by the severe reduction in potential speed and cruising radius inevitably caused by boiler scale.

8. MEASUREMENTS OF WATER IMPURITIES.

(*a*) *Chemical Analysis Method.* The Navy has long recognized the need for constantly checking the feed water impurities and keeping them within low limits. For some years all Naval ships have been provided with the "Navy Standard Boiler Water Testing Apparatus" which is a means for ascertaining the approximate impurity content by chemical analysis. The regulations specify that an analysis of the condensate from each condenser be made every fifteen minutes while underway. It would have been impracticable to have provided Navy ships with an outfit for making a complete water analysis. Even the outfit provided, comprises sixteen items of glassware and four chemicals. When used for determining impurity, the approximate chlorine equivalent is measured and it is assumed that the total impurities are

roughly proportional to the chlorine equivalent. It has, therefore, become customary in the Navy to express the impurity content in terms of grains of chlorine per gallon of water.

In the physical sense there is no chlorine in the water nor can there be, as the vaporization point of chlorine is below the freezing point of water. Even if free chlorine were present it would do little harm except in the presence of oxygen. It should be borne in mind, therefore, that the graduations in grains of chlorine on water impurity meters, are simply arbitrary divisions, and that the term "grains of chlorine," has no meaning in itself. The equivalents of a 1-grain chlorine indication in terms of the various impurities are given in Plate I. It will be noted that a 1-grain indication is equivalent to 2.02 grains of total impurities of which .37 of a grain is classed as harmful and 1.65 grains is classed as harmless.

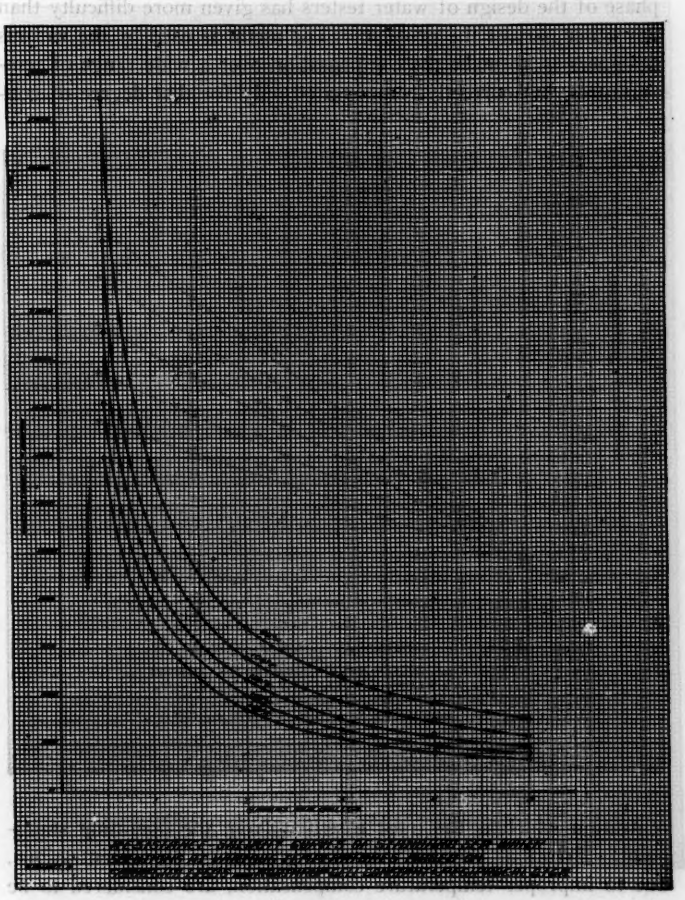
The Navy water testing outfit has served a very good purpose and has undoubtedly brought about a very considerable increase in the efficiency of boiler operation on Navy ships. However, it is somewhat inconvenient and inaccurate in use. It is inconvenient because of the delay and the number of operations required to make a determination, the consequent difficulty of noting the rate of change, the liability of breakage of the glassware, and the necessity for renewing such of the chemicals as are subject to deterioration or replacement of the chemicals which are consumed. It is not very accurate because only a rough determination of total impurity content is possible, errors may be introduced by dirt or deposits in the containers, and lastly because it is dependent upon color reaction which is not sharply defined.

(*b*) *Electrical Conductivity Methods.* Because of the inconvenience and the inadequacies of the chemical method, numerous schemes have been evolved to quantitatively analyze water impurities by electrical conductivity methods. Various titles have been used by the different manufacturers to describe the conductivity method, such as Electrical Salinity Indicators, Concentration Salinity Indicators, Water Testers, and Water Impurity Meters. They one and all, however, are based on the principle of the increase in electrical conductivity of water with the increase of electrolytic impurities, or conversely, the increase in electrical resistance with the decrease of electrolytic impurities.

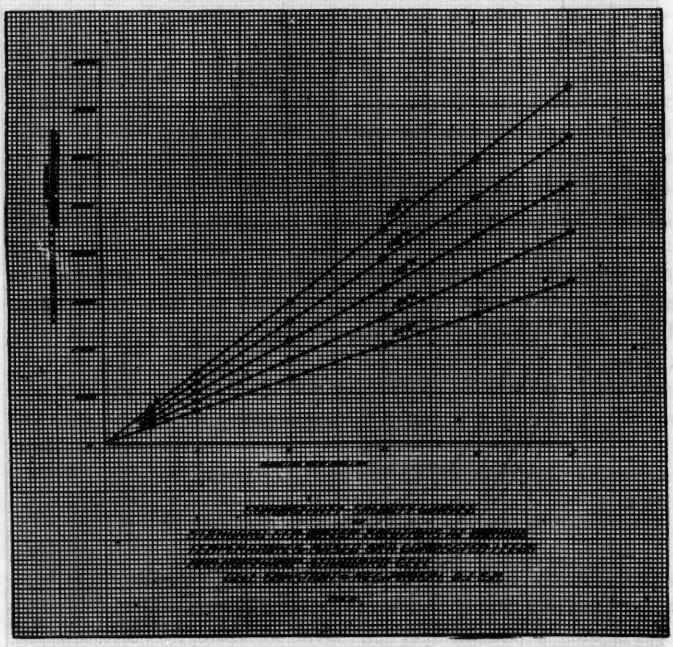
It has been proven by scientists, that the conductivity of absolutely pure water is practically zero. Therefore, any electrolytic impurities increase the conductivity of water to a definite value depending on the character, the amount of the impurities, and the temperature of the solution. Figure 2 shows the characteristic changes in ohmic resistance of dilute sea water electrolyte from ½ to 5 grains per gallon concentration. Figure 3 shows this same data plotted in terms of conductivity. These straight-line conductivity curves further confirm the assumption that the conductivity of pure water is zero. It has been further established, that, provided a solution is very dilute, the conductivity is proportional to the percentage amount of the electrolytic substance (metallic salts), dissolved in the water. Now, while the various electrical water testing schemes are thus able, by measuring conductivity or resistance, to indicate with various degrees of accuracy the percentage amount of any electrolytic substance dissolved in water, they do not, of course, discriminate between one kind of substance and another. Analysis alone can do that. The impurities in boiler water used for marine purposes, are assumed to originate from the sea water, and as these impurities have been ascertained once for all by analysis as being inherent in the source of supply, impurity tests are not made for the purpose of qualitative analysis, but to determine the percentage of the known collective substances in the water.

9. EFFECT OF TEMPERATURE ON THE CONDUCTIVITY OF IMPURE WATER.

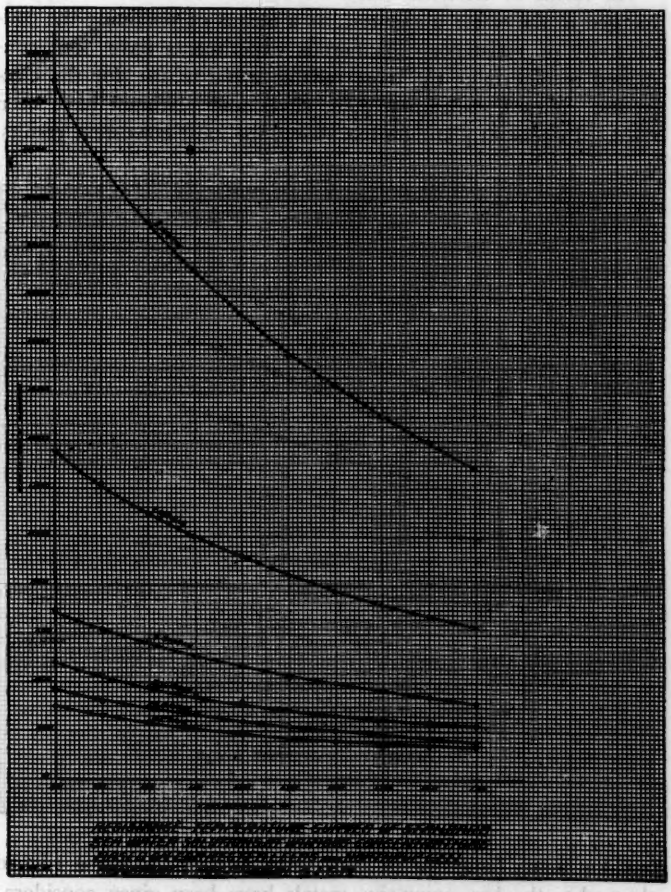
It is unfortunate for the designers of electrical water impurity testers, that the conductivity of electrolyte is not independent of temperature. In order to make the determinations comparable, it is necessary to evolve some means of reducing all conductivity or resistance readings to some standard temperature. The resistance of impure water decreases approximately 1.2 per cent with each degree Fahrenheit increase in temperature. This means that if the calibration of a conductivity or resistance meter were to be correct at 70 degrees F., it would have an error of 120 per cent at 170 degrees F. Unfortunately, also, the temperature effect varies considerably with each degree of concentration, so that in order



to have an accurate compensation to a standard temperature, it would be necessary to work out and apply a series of compensating variable resistances, one for each degree of concentration. This phase of the design of water testers has given more difficulty than any other part of the problem. The methods used by the various designers is, to make the compensation over a wide range of temperatures, but correctly compensated only for that degree of con-

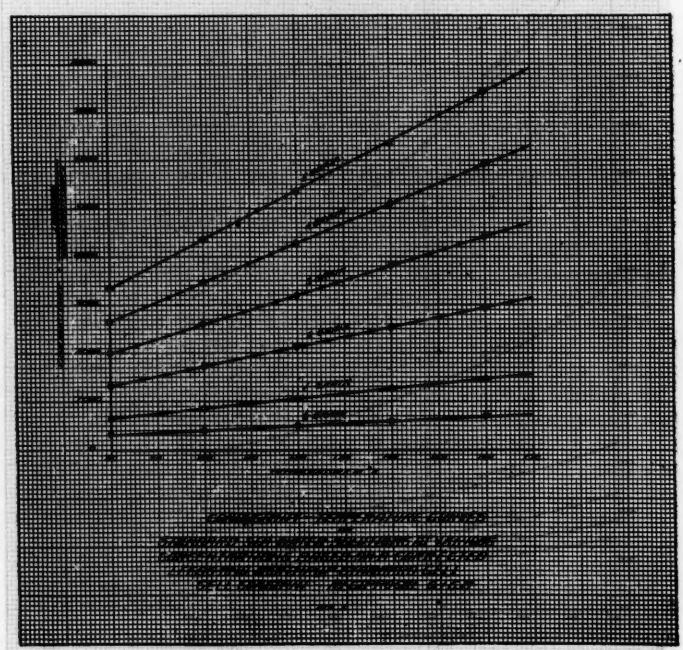


centration which is the most essential from the operating standpoint. Errors at very low or very high degree of concentrations, due to improper temperature compensation, are considered to be of less disadvantage than a complicated fully compensated system. Figure 4 shows the characteristic changes in resistance, due to varying temperature, and Figure 5 shows the same data plotted in terms of conductivity.



10. THE EFFECT OF CURRENT ON THE ELECTRODES OF CONDUCTIVITY CELLS.

All forms of electrical water testers use metal electrodes submerged in the water under test, as a means of obtaining the conductivity or resistance. Numerous experiments have been conducted to determine the most reliable metal to use for the electrodes. Nickel, palladium, gold, silver and platinum have been tried. In general platinum has shown the most consistent results,



but due to its high cost in solid form sufficient to make rugged electrodes, the less expensive metals have been given consideration. For continuous operation, it is necessary to use alternating current, for the reason that if direct current were to be used, there would be continuous electrolytic deposition of the metal from the positive electrode on to the negative electrode of the conductivity

cell. Even with the use of alternating current there is a continuous formation of occluded gas on the electrodes formed by the decomposition of the water, which if not relieved, would gradually build up an almost invisible but high resistance envelope around the electrodes. The relief from the occluded gas is accomplished by designing the cell so that the water may freely circulate around the electrodes, by using low current density, and also by employing a electrolytic deposit of platinum black on the electrodes. Platinum black deposit is obtained by immersing the electrodes in a solution containing 3 per cent platinum chloride and 1/40 per cent lead acetate in water and then passing a small direct current between the electrodes, first in one direction for a few minutes and then in the reverse direction. Platinum black is very efficient in maintaining the constancy of resistance by reducing or preventing the formation of occluded gas by virtue of its catalytic action.

11. EFFECT OF VARIATIONS IN ELECTRODE METAL GUARDS.

The effective resistance of salinity cells has been found to be influenced to a considerable extent by the presence of a dirty or greasy coating on the inner side of the metal protective cap or guard. These guards, although used merely for mechanical protection of the electrodes, when clean, act to some extent as a shunt path for the current, thereby lowering the effective resistance of the cell. Errors from this source are shown from the following test figures using a standard double electrode cell in approximately a 2-grain electrolyte at 85 degrees F.

Cell with clean guard	=	815 ohms
Cell with guard removed	=	845 ohms
Cell with dirty guard	=	865 ohms

In other words, the lowest cell resistance is obtained by the use of a clean metal guard which, in conjunction with the electrolyte contained in the guard, serves as a partial shunt path for the current. When the guard is dirty such as is caused by oxides, grease, or other insulating deposits obtained in service, the effective cell resistance is increased approximately 6 per cent. From the above analysis, it is considered desirable for uniformity of service results, to coat the inside of all metal protective guards, with a permanent insulation, thereby eliminating the unknown

errors resulting from various degrees of service deposits on the guards. No appreciable effect was found in the cell resistance by interchanging different clean cell guards, or by interchanging different cell guards which were insulated.

12. STANDARDIZATION OF ELECTRODE AREA AND SEPARATION. (CELL CONSTANT.)

In determining the resistance or conductivity of electrolytic solutions, it is necessary for comparison of data to establish a standard cell constant. The resistance offered to a flow of electric current by a given electrolytic cell containing an electrolyte, varies inversely with the area of the electrodes, and directly with the effective distance separating them. As an illustration, if the resistance of a cell with electrodes of unit area 1×1, and with a separation of 1, has one unit resistance, a cell with electrode area 2×2 would have $\frac{1}{4}$ unit resistance at 1 unit separation. But with a separation of 2, it would have $2 \times \frac{1}{4}$, or $\frac{1}{2}$ unit resistance. On this basis, a cell of 2 cube units has $\frac{1}{2}$ the resistance of a cell with 1 cube unit, and therefore cell resistances vary as the reciprocal of its cube dimensions. The effect of these factors is expressed as the cell constant C. A great many finite standard units have been used by experimenters, but the value accepted for Naval standardization is on the basis of 0.1 centimeter cube, for feed-water impurities.

The cell constant on the above basis is expressed for practical use as the reciprocal 0.1 cm. The standard cell used in plotting the curves shown in Figure 2, has electrodes each 1.6-centimeter area on one side. The electrode separation (X), may be calculated for this cell as follows:

$$C = 0.1 \text{ cm. or } 0.1 = \frac{X}{1.6^2}, \text{ in which } X = 0.256 \text{ centimeters.}$$

If the resistance values of an electrolytic solution were to be taken with a cell in which $C = 0.2$ cm, the data would be comparable if the results were to be multiplied by the factor 2.

In using standard cells made up of the above dimensions, it has been found impracticable to adjust the electrode separation exactly, due to the frailness of the platinum electrodes used. It

is therefore, the practice to set the electrodes approximately standard, and then obtain a resistance reading using a 1/50 normal potassium chloride (KCL) solution, the specific conductivity of which has been accurately determined by scientists, at various temperatures but in terms of chemically pure water in the solution. The cell resistance in this solution, and the resistance of the water used in making the solution, is measured by means of a 60-cycle alternating current Wheatstone bridge. The formula for calculating the error of the standard cell and the derivation of the formula is shown as follows:

$$C = \frac{\text{Specific conductance}}{\text{Cell conductance}}$$

We designate the factors entering into the above formula as follows:

C = cell constant

L = specific conductance of pure 1/50 N.KCL

Ls = specific conductance of the test 1/50 N.KCL

Lw = specific conductance of water used in 1/50 N.KCL

Rs = ohms cell resistance in test 1/50 N.KCL

Rw = ohms cell resistance in water used in test 1/50 N.KCL

Conductance = reciprocal of resistance

Specific conductance = cell constant divided by resistance

Total specific conductance = the sum of the specific conductance of its constituents.

Arranging the basic formula and solving in terms of the known factors L, Rw and Rs.

$$C = \frac{Ls}{\frac{1}{Rs}} \quad \text{Substituting } Ls = L + Lw$$

$$C = \frac{L + Lw}{\frac{1}{Rs}} \quad \text{Substituting } Lw = \frac{C}{Rw}$$

$$C = \frac{L + \frac{C}{Rw}}{\frac{1}{Rs}} \quad \text{Substituting } C = \frac{Ls}{\frac{1}{Rs}}$$

$$\therefore C = \frac{L + \frac{\frac{Ls}{1}}{Rw}}{\frac{1}{Rs}} = (L.\ Rs + Ls.\ Rs)\ \frac{Rs}{Rw}$$

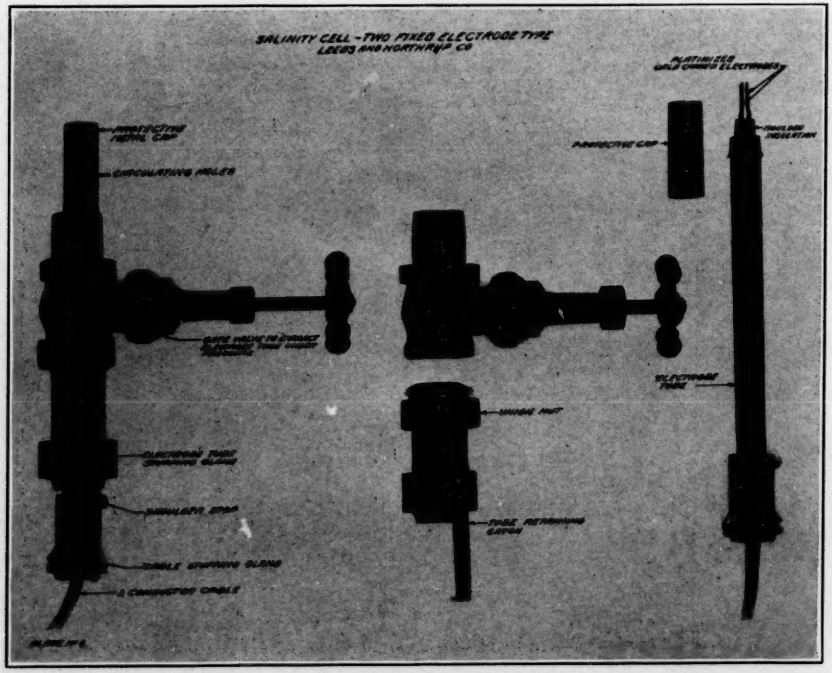
Since Ls = L nearly, we can write, if magnitudes of the second order are neglected:

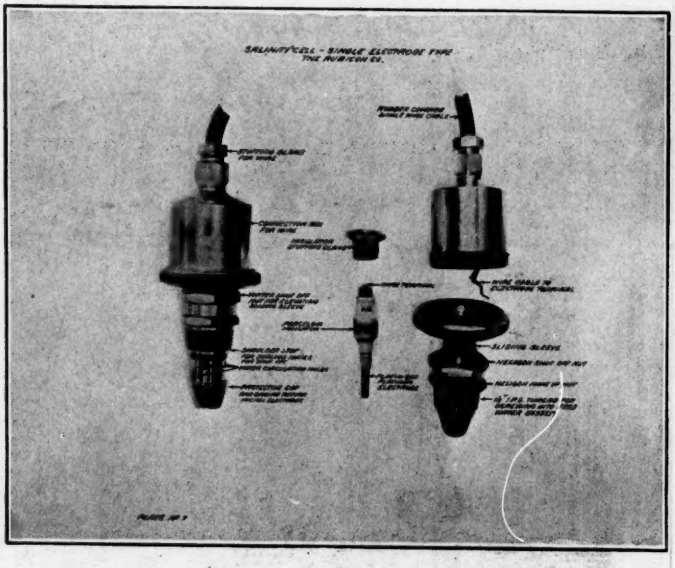
$$C = (L.\ Rw + L.\ Rs)\ \frac{Rs}{Rw}$$

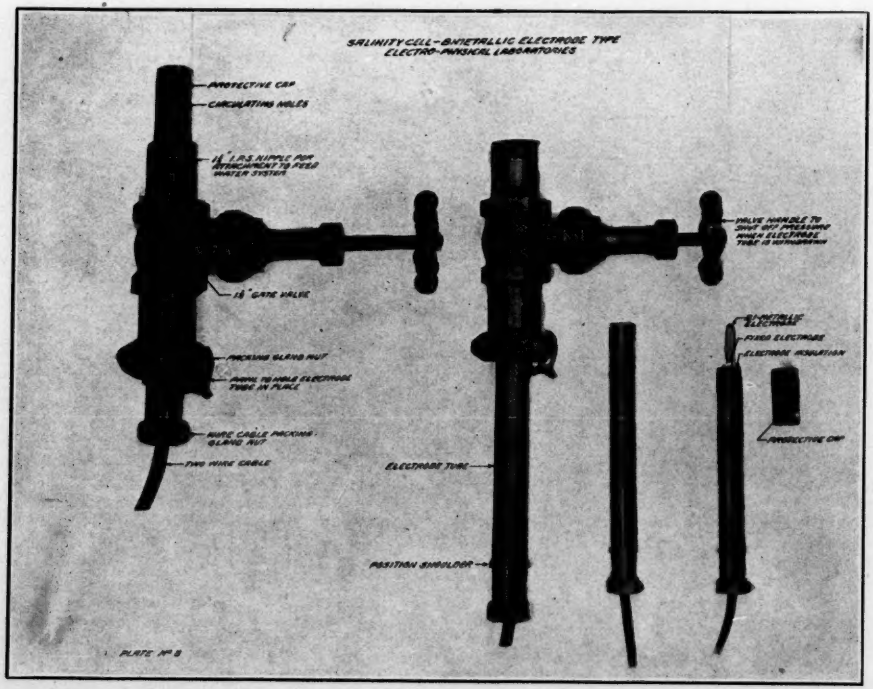
The specific conductivity of 1/50 N.KCL at various temperatures have been determined by experimenters as follows:

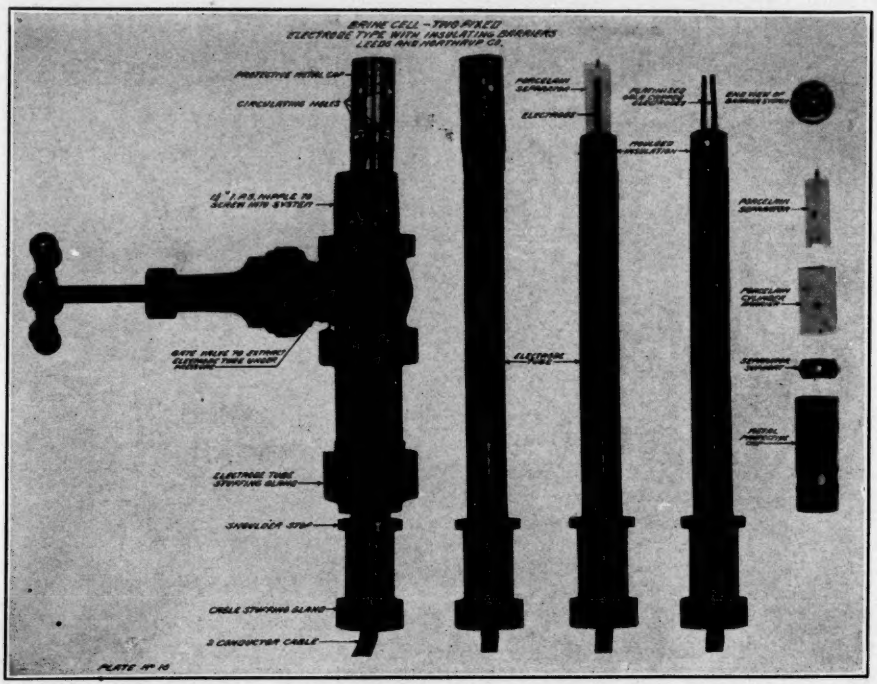
Temperature C. Degrees	KCL 1/50 Normal
20	.002501
21	.002553
22	.002606
23	.002659
24	.002712
25	.002765
26	.002819
27	.002873
28	.002927
29	.002981
30	.003036

The variation of the standard cell as checked by the research standard method, is expressed in per cent, and all data taken with the working standard in determining the concentration of the various test solutions of sea water are corrected on this basis.


SALINITY CELL - TWO FIXED ELECTRODE TYPE
LEEDS AND NORTHRUP CO
PLATINIZED
GOLD COATED ELECTRODES
PROTECTIVE
METAL CAP
CIRCULATING HOLES
PROTECTIVE CAP
TAPERED
INSULATOR
GLASS TUBE
SUPPORTING GLAND
ELECTRODE
TUBE
UNION NUT
GLASS
SUPPORTING GLAND
TUBE RETAINING
CAPTIVE
RUBBER STOPPER
CONNECTING CABLE
PLATE No 6


SALINITY CELL - SINGLE ELECTRODE TYPE
THE RUBICON CO.


SALINITY CELL - BIMETALLIC ELECTRODE TYPE
ELECTRO-PHYSICAL LABORATORIES
PROTECTIVE CAP
CIRCULATING HOLES
1/4" I.P.S. NIPPLE FOR ATTACHMENT TO FEED WATER SYSTEM
1/4" GATE VALVE
PACKING GLAND NUT
HOLE TO HOLD ELECTRODE TUBE IN PLACE
WIRE CABLE PACKING GLAND NUT
TWO WIRE CABLE
ELECTRODE TUBE
POSITION SHROUDOR
VALVE HANDLE TO SHUT OFF PRESSURE WHEN ELECTRODE TUBE IS WITHDRAWN
BIMETALLIC ELEMENT
FIXED ELECTRODE
ELECTRODE INSULATION
PROTECTIVE CAP
PLATE No 8


BRINE CELL – TWO FIXED
ELECTRODE TYPE WITH INSULATING BARRIERS
LEEDS AND NORTHRUP CO.
PROTECTIVE METAL CAP
CIRCULATING HOLES
4" I.P.S. NIPPLE 28 SCREW INTO 8-WAY
GATE VALVE TO EXTRACT ELECTRODE TUBE UNDER
ELECTRODE TUBE STUFFING GLAND
SHOULDER STOP
CABLE STUFFING GLAND
2 CONDUCTOR CABLE
PORCELAIN SEPARATOR
ELECTRODE
PLATINIZED PLATINUM ELECTRODE
ENLARGED INSULATION
ELECTRODE CABLE
END VIEW OF BARRIER STEM
PORCELAIN SEPARATOR
PORCELAIN CYLINDRICAL BARRIER
SCREWDRIVER SLOTTED
METAL PROTECTOR
PLATE Nº 10

13. CELL CONSTRUCTION.

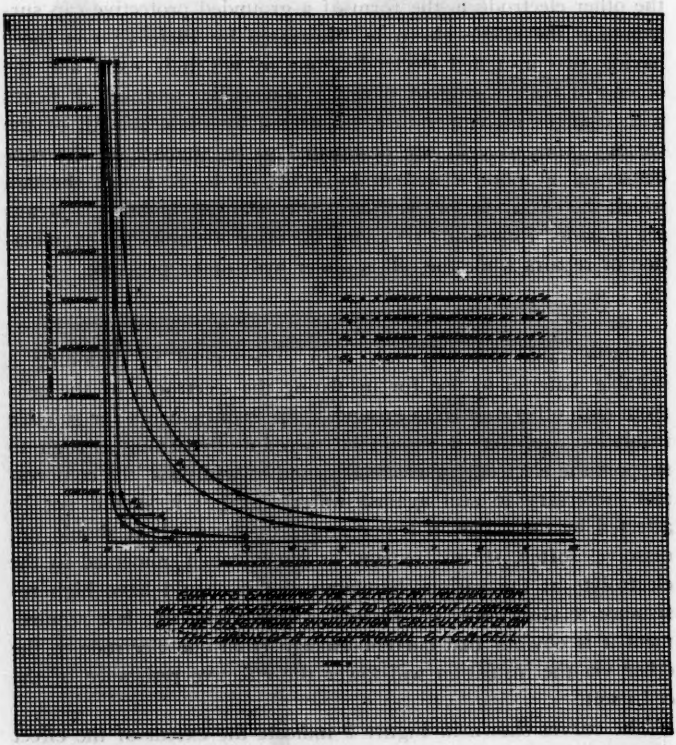
The salinity and brine cells used in the Navy are of three general types as regards to the electrodes; viz.: (1) those having two fixed electrodes each insulated from ground, illustrated by Plate 6; (2) those having one fixed electrode insulated from ground and the other electrode in the form of a grounded protective cap surrounding the single electrode, illustrated by Plate 7; (3) those having two insulated electrodes, one of which is fixed and the other (or both) being of movable bi-metallic metal which flexes with changes in temperature, illustrated by Plate 8. All of the various types of cells are designated to be attached to the condenser or evaporator system by a 1¼ I. P. S. nipple through which the electrodes project into the condensate or brine. Means are provided for readily removing the electrode units for examination or repairs while the cell is under hydrostatic pressure.

14. ELECTRODE INSULATION.

Considerable difficulty has been experienced in obtaining a suitable material to use for insulating the electrodes of the cells from each other and from ground. The requirements for suitable insulation are, that it should be mechanically strong enough to support the electrode or electrodes, and to withstand the compression necessary to seal against water leakage; it should not be affected by the range in temperatures experienced; it should not dissolve, soften, shrink, expand or crack, when immersed in the electrolyte at the various temperatures; it should have high resistance to absorption of moisture. Concerning this last requirement it is obvious that current leakage from one electrode to the other via the insulation will lower the effective resistance of the cell itself.

The curves shown in Figure 9 indicate the extent of the effect of insulation current leakages on the cell readings. From these curves it will be seen that for condensates of ½ grain impurity or less, in order to reduce the errors from this source to 2 per cent or less, the electrode insulation should not be less than approximately 200,000 ohms. For impurities of 5 grains or more, the errors from electrode leakages are not so disturbing, due to the increasing conductivity of the electrolyte.

Various materials have been tried out for electrode insulation, such as porcelain, and synthetic moulded materials using different binders. The most successful material found and which has given excellent results, is made of bakelite with an asbestos binder. The insulation resistance of this material after a long period of



use is of the order of 100 megohms. In order to check the insulation resistance of a cell in service, it is usual to remove the cell from the condensate, dry off all surface moisture and then measure the insulation between electrodes, and between the disconnected line leads, by means of an ohmmeter or other suitable insulation resistance appliance.

15. THE BRINE CELL.

The foregoing description of the electrolytic cells has been on the basis of cells used for water impurities. These cells have been designed for best operation on comparatively weak electrolytic solutions not exceeding the equivalent of 10 grains of chlorine impurity per gallon. In order to obtain ohmic resistance values suitable for the best and the most accurate operation of the Wheatstone bridge systems and other systems used as indicators, a cell constant of reciprocal 0.1 cm. is generally used and this cell electrode factor has been generally accepted as standard for *impurity* indicators.

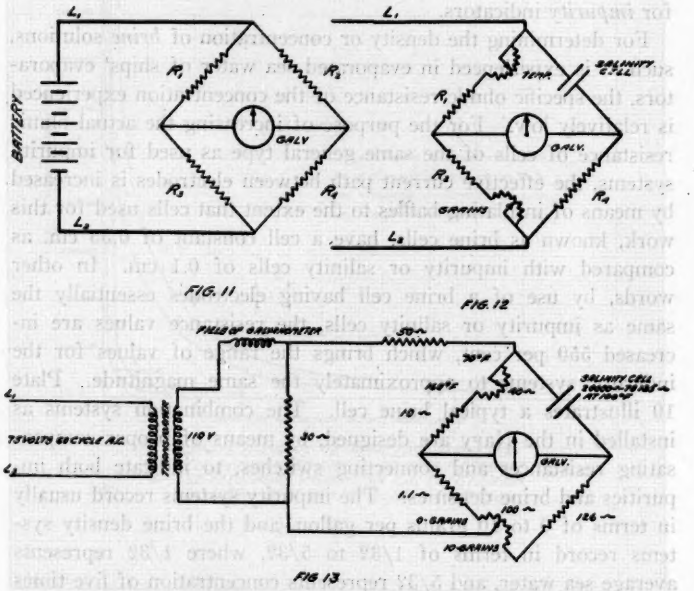
For determining the density or concentration of *brine* solutions, such as is experienced in evaporated sea water of ships' evaporators, the specific ohmic resistance of the concentration experienced is relatively low. For the purpose of increasing the actual ohmic resistance of cells of the same general type as used for impurity systems, the effective current path between electrodes is increased by means of insulating baffles to the extent that cells used for this work, known as brine cells, have a cell constant of 0.55 cm. as compared with impurity or salinity cells of 0.1 cm. In other words, by use of a brine cell having electrodes essentially the same as impurity or salinity cells, the resistance values are increased 550 per cent, which brings the range of values for the indicator systems to approximately the same magnitude. Plate 10 illustrates a typical brine cell. The combination systems as installed in the Navy are designed, by means of proper compensating resistances and connecting switches, to indicate both impurities and brine densities. The impurity systems record usually in terms of 0 to 10 grains per gallon, and the brine density systems record in terms of 1/32 to 5/32, where 1/32 represents average sea water, and 5/32 represents concentration of five times that of average sea water.

16. DETAILED DESCRIPTION OF VARIOUS ELECTRICAL SYSTEMS.

The Rubicon Co. Salinity Indicator Systems. The Rubicon Company has designed for Navy installations, two general types of salinity or impurity indicators, based on the electrical conductivity principle. Both types employ the same design of salinity

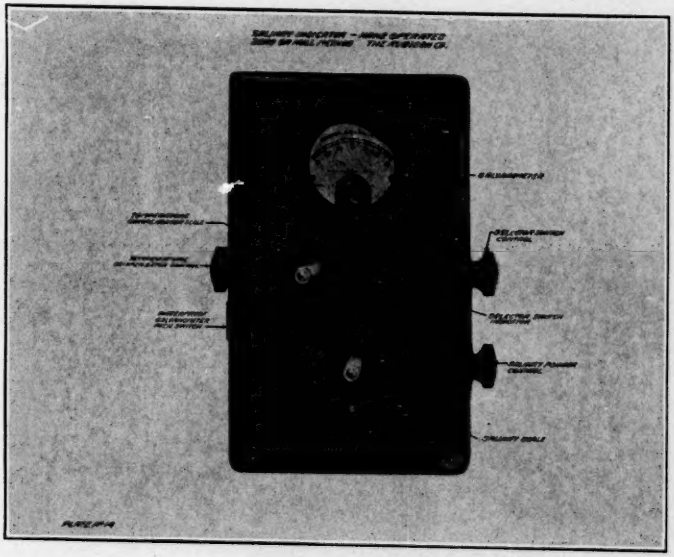
cell, but use different types of indicators for the purpose of obtaining simplicity in one case, and automatic warning signals in the other case. A description and explanation of the working principles is furnished as follows:

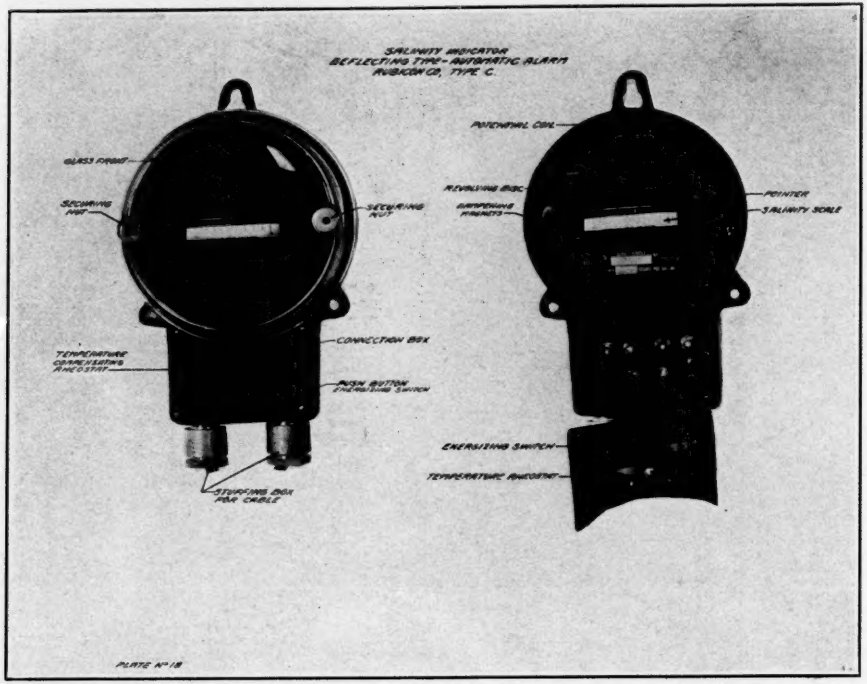
Type C, Wheatstone Bridge, Zero or Null Method. This system is built around the Wheatstone bridge principle in which, referring to Figure 11, when the ohmic resistance of $\frac{R1}{R2} = \frac{R3}{R4}$ the galvanometer is balanced to zero deflection. The application



SALINITY INDICATOR—ZERO OR NULL METHOD. THE RUBICON CO.—TYPE C.

of this principle to the salinity indicator is shown in the fundamental sketch, Figure 12. The complete application is shown in Figure 13 which places the line transformer, the temperature compensating rheostat, the salinity indicating rheostat, and the salinity cell. The object of the transformer is not only to furnish




SALINITY INDICATOR
DEFLECTING TYPE - AUTOMATIC ALARM
AMBROM CO. TYPE C.
GLASS FRONT
SECURING NUT
SECURING NUT
TEMPERATURE COMPENSATING THERMOSTAT
CONNECTION BOX
PUSH BUTTON
STUFFING BOX FOR CABLE
POTENTIAL COIL
REVOLVING MAGNETS
POINTER
SALINITY SCALE
EXCITING SWITCH
TEMPERATURE RHEOSTAT
PLATE No 18

a suitable ratio of A. C. volts to the system, but is primarily for the purpose of completely insulating the power supply from the indicator and cell, thus avoiding the possibility of cross currents from power line grounds. Plate 14 shows the complete assembled indicator of this system. To obtain an indication of salinity, it is necessary to first set the temperature compensating rheostat to the marked value, approximating the temperature of the feed water in which the cell is located, and then adjusting the salinity rheostat until the galvanometer shows no deflection. At this setting, the grains of salinity per gallon is observed from the calibrated dial of the salinity rheostat.

Type Cl, Deflecting Type With Automatic Alarm Signals. This system was evolved to furnish an automatic alarm system in combination with an indicator which is calibrated to deflect continuous indications of the degree of salinity. The zero or null method is fundamentally a more accurate method, but does not readily lend itself to automatic signals or automatic indications of salinity. The principle of operation of the type C indicator is based on the principle of an A. C. induction watt-hour meter, as shown in Figure 15. In this type of meter, the metal disc is caused to rotate by inductive reaction produced by the current and voltage coils somewhat similar in principle to an induction motor. Figure 16 shows the adaptation of this principle to a salinity indicator by placing the cell in series with one of the current coils, while the other current coil is constantly excited from the line voltage. The disc is restrained from rotating by a counter torque spring. The constantly excited current coil deflects the disc in one direction against the torque of the spring, and thus with the cell circuit open or with infinite resistance at the cell, the minimum or zero indication on the meter is established. With the cell circuit closed and with current flowing across the cell torque is produced in the one-current coil of the meter in opposition to the torque produced by the constantly excited current coil. Thus the resulting deflection is calibrated in terms of the salinity conductivity, or, in terms of grains per gallon of impurities. The completed circuit showing the transformer, potential coil, temperature rheostat, salinity cell and signal light, is shown in Figure 17. The signal lights are arranged so that a green light is " on" when the signal circuit is energized. When the disc deflects to the low size

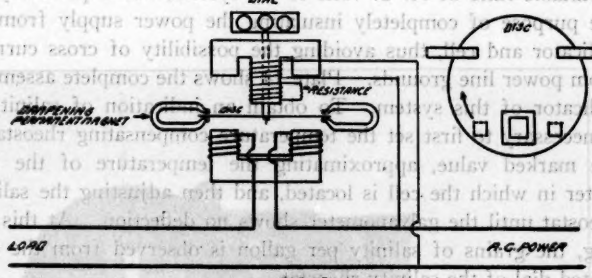


FIG. 15

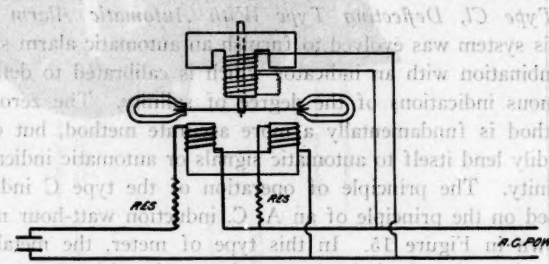


FIG. 16

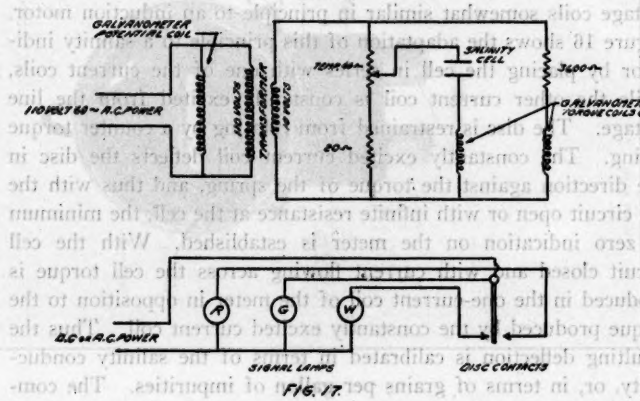


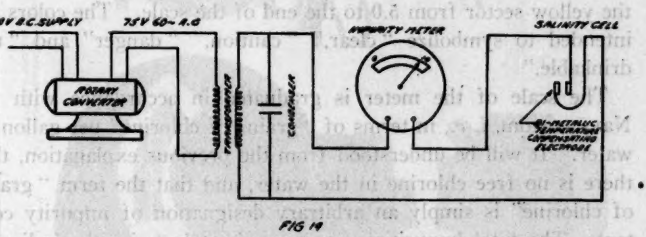
FIG. 17.

SALINITY INDICATOR AND SIGNAL LIGHT CONTROLLER TYPE C., THE RUBICON CO.

or minimum impurities, a contact on the disc establishes a white light. When the disc deflects to high side or maximum impurities, a contact on the opposite side of the disc establishes a red light. This system, excluding the hand-operated temperature compensating rheostat, is designed to give automatic and continuous indications of the degree of salinity with warning signal of dangerous concentrations. Plate 18 shows an assembly, and an exposed view of the indicator and the hand-temperature compensator.

The Ruben Water Impurity Meter System. The Ruben water impurity meter system was developed by the Electro-Physical Laboratories to fill the need for a simple, reliable, continuously automatic indicating system, with automatic temperature compensation.

The elementary principle of the Ruben system is shown by Figure 19. It will be seen that alternating current of 60 cycles in



THE RUBEN IMPURITY METER SYSTEM—ELECTRO-PHYSICAL LABORATORIES.

a straight series circuit is employed. With an ordinary meter, this would not enable accurate measurement at all parts of the scale. The latest design of meter employed in this case has its moving iron element specially designed so as to progressively reduce the angular deflection with increasing current, the effect of which is to give higher accuracy and readability at the low concentrations of salinity and making it possible to read with serviceable accuracy up to 10 grains.

The potential applied across the cell is about 25 volts, which is relatively high, thus reducing the effect of gas occlusion. This

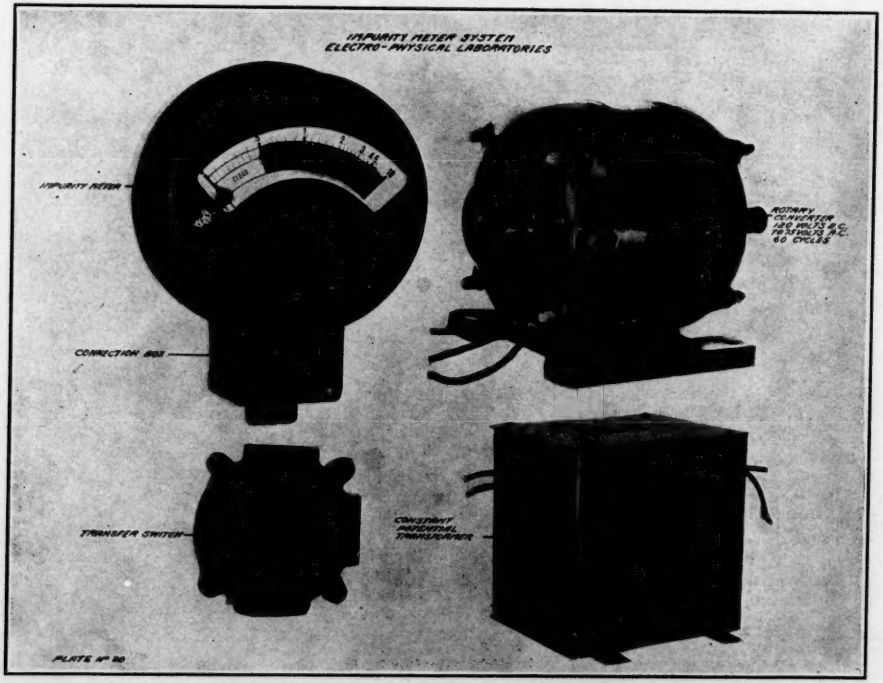
makes it possible to consider the use of plain platinum electrodes, and so avoid the use of black platinized surfaces which may wear off and require renewal. However, in the latest development, platinized surfaces are used. The potential is held constant by using a constant voltage transformer especially developed for the purpose. The transformer has inherent characteristics such that it will maintain a fairly constant potential on the out-put side, with variations of the ship's supply between 100 to 125 volts.

The Meter. The indicating meter used is shown in Plate 20 along with the rotary converter, transformer and switch, and is of the deflecting type similar in principle to the usual moving iron type A. C. voltmeters and ammeters which have an exciting coil but no permanent magnets. The usual mechanical dampening vanes are provided in the meter to reduce the oscillations of the pointer. The dial of the meter is divided into color sectors. The white sector extends from 0 to 0.5 grains, the green sector from 0.5 to 1.0 grains, the red sector from 1.0 to 5.0 grains, and the yellow sector from 5.0 to the end of the scale. The colors are intended to symbolize "clear," "caution," "danger" and "undrinkable."

The scale of the meter is graduated in accordance with the Navy custom, *i. e.*, in terms of "grains of chlorine" per gallon of water. It will be understood from the previous explanation, that there is no free chlorine in the water, and that the term "grains of chlorine" is simply an arbitrary designation of impurity content. The total impurities are approximately twice the indicated "grains of chlorine," and the harmful impurities are on the average a little more than ⅓ of the value of the indicated "grains of chlorine."

The Cell. The removable cell with its outer casing and gate valve is shown in Plate 8. Two electrodes are used, the active parts of which consist of a platinum button ⅜ inch in diameter. One of these buttons is mounted on a short rigid support, the other is mounted at the end of a U-shaped bi-metallic strip. Consideration has been given to using bi-metallic metal for both electrodes. All exposed parts, except the button, are covered with several coats of a flexible insulating material.

The cell electrodes are in series with the meter, and with a source of current of constant potential, the indications of the


IMPURITY METER SYSTEM
ELECTRO-PHYSICAL LABORATORIES
IMPURITY METER
CORRECTION BOX
TRANSFER SWITCH
ROTARY CONVERTER 120 VOLTS D.C. TO 110 VOLTS A.C. 60 CYCLES
CONSTANT POTENTIAL TRANSFORMER
PLATE Nº 20

SALINITY INDICATOR - ZERO OR NULL METHOD
WALL TYPE - MANUALLY OPERATED
LEEDS AND NORTHRUP CO.

GALVANOMETER
HAIR LINE INDEX
SALINITY REVOLVING SCALE
SALINITY RHEOSTAT CONTROL HANDLE
SNAP SWITCH FOR ENERGIZING SYSTEM
TEMPERATURE COMPENSATING BULB AND CONTROL
BRINE CELL TURN KEY FOR BARE ELECTRODES
GALVANOMETER PUSH SWITCH

PLATE No 23

meter are governed by the conductivity of the water between the electrodes of the cell, and the length of the path between the electrodes. The conductivity of the water is a function of its impurity content and its temperature. The bi-metallic strip is designed so that when the temperature increases the conductivity of the water, it widens the gap between the electrodes by virtue of its deforming characteristics. The cell is designed so that at normal concentration the increase in the gap compensates for the increased conductivity due to the rise in temperature. The advantages claimed by the bi-metallic eliminates the usually manually operated temperature compensating rheostat. However, in common with other systems, the compensation can be correct for only one point in the range of impurities. This is for the reason that the temperature coefficient of conductivity varies with the concentration, while the coefficient of the bi-metallic strip or a compensating rheostat, stays fixed.

The Leeds and Northrup Company Concentration Salinity Indicator Systems. The Leeds and Northrup Company have three general types of salinity, impurity or concentration indicators in Naval Service, based on the conductivity principle. All types employ the same design of double electrode cells, but use different types of indicators for the purpose of obtaining simplicity in one case, and automatic indications and warning signals in the other cases. A description and explanation of the working principles are furnished as follows:

Wall Type Manually Operated Concentration Indicator. This system is built around the Wheatstone bridge principle, and uses the zero or null method. Referring to Figure 21, when the resistances AD = BC, the galvanometer shows no deflection and the cell

$$\text{resistance } C = \frac{AD}{B}$$

The complete application of the Wheatstone bridge principle, is shown in Figure 22 which places the line transformer, the temperature compensating rheostat, the salinity indicating rheostat and the salinity cell. Plate 23 shows the complete assembled indicator of this system. To obtain the indication of salinity, it is necessary to first set the temperature compensating rheostat to the marked

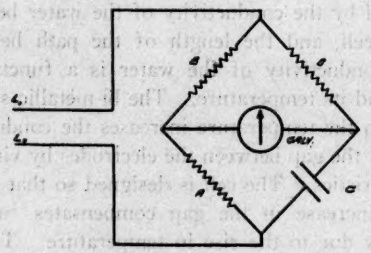


FIG. 21

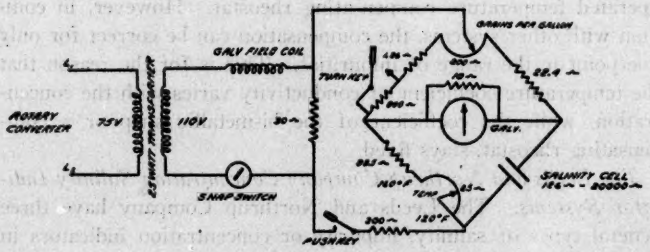


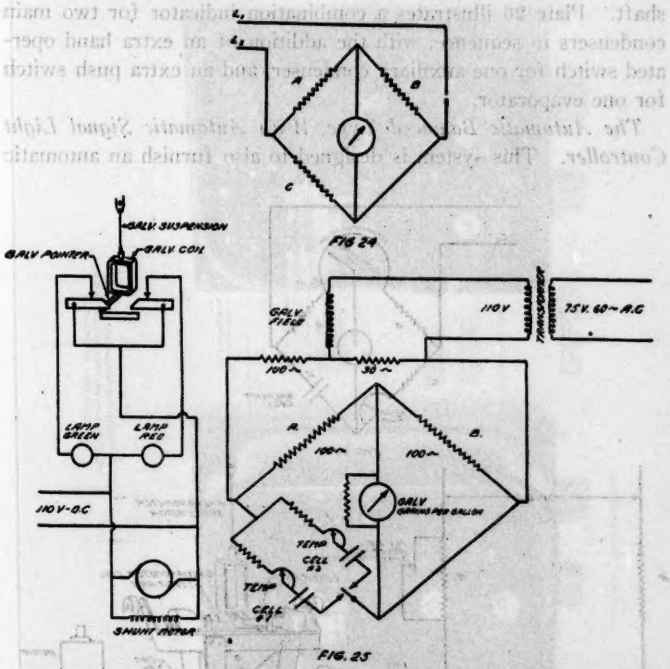
FIG. 22

LEEDS AND NORTHRUP WALL TYPE MANUALLY OPERATED CONCENTRATION INDICATOR ZERO OR NULL METHOD.

value approximating the temperature of the condensate in which the cell is located, and then adjusting the salinity rheostat until the galvanometer shows no deflection. At this setting, the grains of salinity per gallon is observed from the calibrated dial of the salinity rheostat.

The Deflecting Type, With Automatic Signal Light Controller. This system is designed to furnish an automatic alarm light, in combination with a deflecting galvanometer calibrated to deflect continuous indications of the degree of salinity. The fundamental difference between the indicating part of this type and the wall type described, is that the Wheatstone bridge system is left unbalanced by the absence of the variable salinity rheostat as shown fundamentally in Figure 24. The amount of unbalance of the bridge system, which occurs as the cell resistance changes with the degree of salinity, is calibrated on the dial scale of the galva-

nometer, in terms of grains of chlorine per gallon. In this system as shown in the developed diagram Figure 24, a hand-operated temperature compensating rheostat is used in one leg of the bridge which is manually adjusted to the cell temperature prior to observing the galvanometer deflection.



LEEDS AND NORTHRUP DEFLECTING TYPE CONCENTRATION. SALINITY INDICATOR WITH SIGNAL ALARM.

The automatic signal light feature of the indicator is operated by means of a motor-driven cam shaft on which are mounted two or more cams, depending on the number of cells in the system. These cams in sequence, operate a system of rocker arms and levers which engage with the pointer of the galvanometer. When the pointer deflects to the low side of the scale, it enables one of

the levers to close a contact which in turn flashes a green signal light. When the pointer deflects to the high side of the scale, the corresponding action on a second lever flashes a red signal light. By means of the rotating cams, salinity indications of the various cells and warning signals are registered in sequence at time intervals, depending on the speed of the motor-driven cam shaft. Plate 26 illustrates a combination indicator for two main condensers in sequence, with the addition of an extra hand operated switch for one auxiliary condenser, and an extra push switch for one evaporator.

The Automatic Balanced Type, With Automatic Signal Light Controller. This system is designed to also furnish an automatic

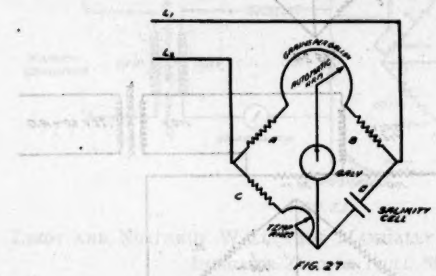


FIG. 27

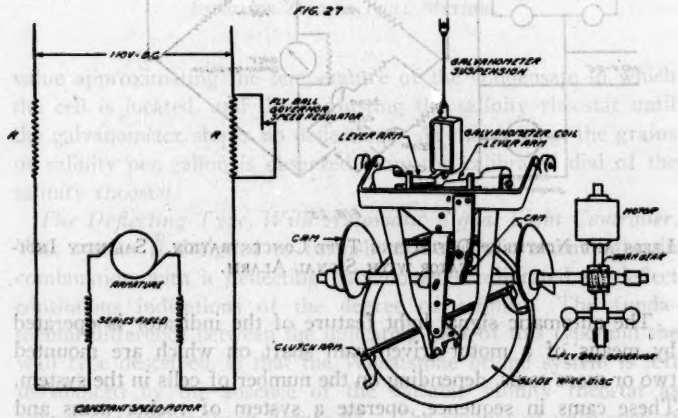
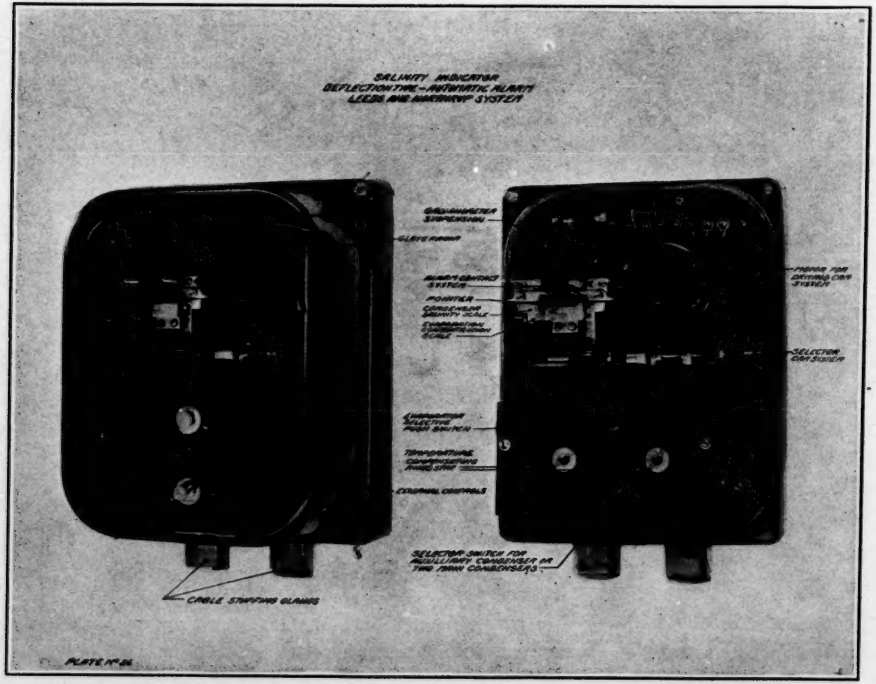
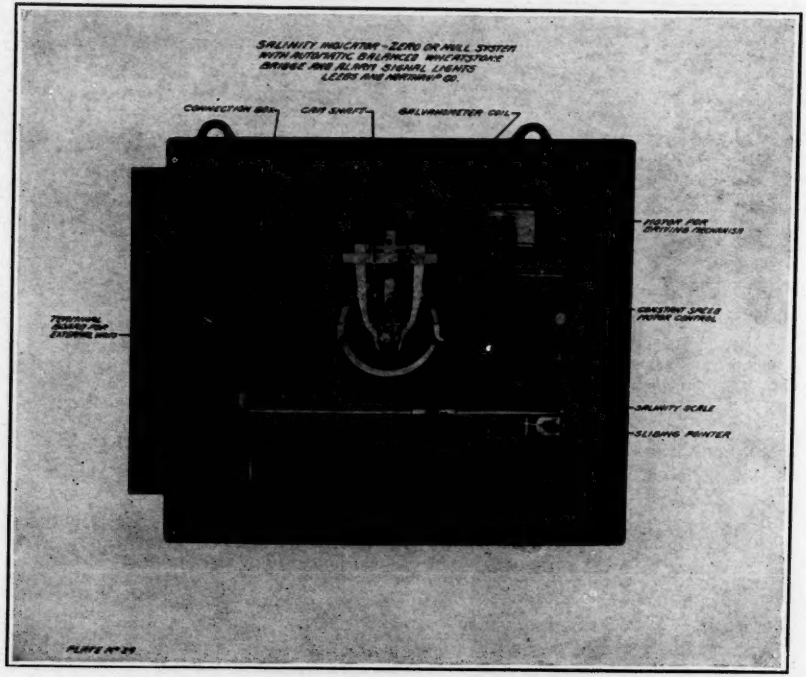


FIG. 28

LEEDS AND NORTHRUP CONCENTRATION SALINITY INDICATOR—AUTOMATIC BALANCED TYPE WITH AUTOMATIC SIGNAL LIGHTS.


SALINITY INDICATOR
DEFLECTION TYPE — AUTOMATIC ALARM
LEEDS AND NORTHRUP SYSTEM
GALVANOMETER SUSPENSION
GLASS FRONT
ALARM CONTACT SYSTEM
POINTER
CONDUCTANCE SALINITY SCALE
TEMPERATURE CONDUCTANCE SCALE
TEMPERATURE SELECTOR PLUG BUTTON
TEMPERATURE CORRECTION PLUG STRIP
CONTROL KNOBS
CABLE STUFFING GLANDS
SELECTOR SWITCH FOR RESISTANCE COILS OR TWO THIN CONDENSERS
TERMINALS FOR OUTSIDE CIRCUITS
SELECTOR CAM SHAFT
PLATE No. 24


SALINITY INDICATOR - ZERO OR NULL SYSTEM
WITH AUTOMATIC BALANCED WHEATSTONE
BRIDGE AND ALARM SIGNAL LIGHTS
LEEDS AND NORTHRUP CO.
CONNECTION BOX
CAM SHAFT
GALVANOMETER COIL
MOTOR FOR DRIVING RHEOSTATS
CONSTANT SPEED MOTOR CONTROL
SALINITY SCALE
SLIDING POINTER
PLATE No 29

alarm light, but in combination with an automatic balanced Wheatstone bridge. In this system the salinity is registered by the movement of a slide wire in the bridge circuit, as shown in Figure 27, which is kept in automatic balance by means of a motor-driven mechanism, as shown in Figure 28. When the galvanometer pointer is deflected from its normal zero position, it permits a systems of levers, clutch and cams to operate the slide wire bridge balancing rheostat, the torque required being supplied by an automatic constant speed motor. The adjusting movement of the slide wire continues until the galvanometer pointer reached its zero position. The salinity indications are recorded by the calibrated position of the rheostat slide arm. The signal light contact disc is connected to the disc carrying the slide resistance and moves with it. Plate 29 illustrates one form of indicator based on the above-described system.

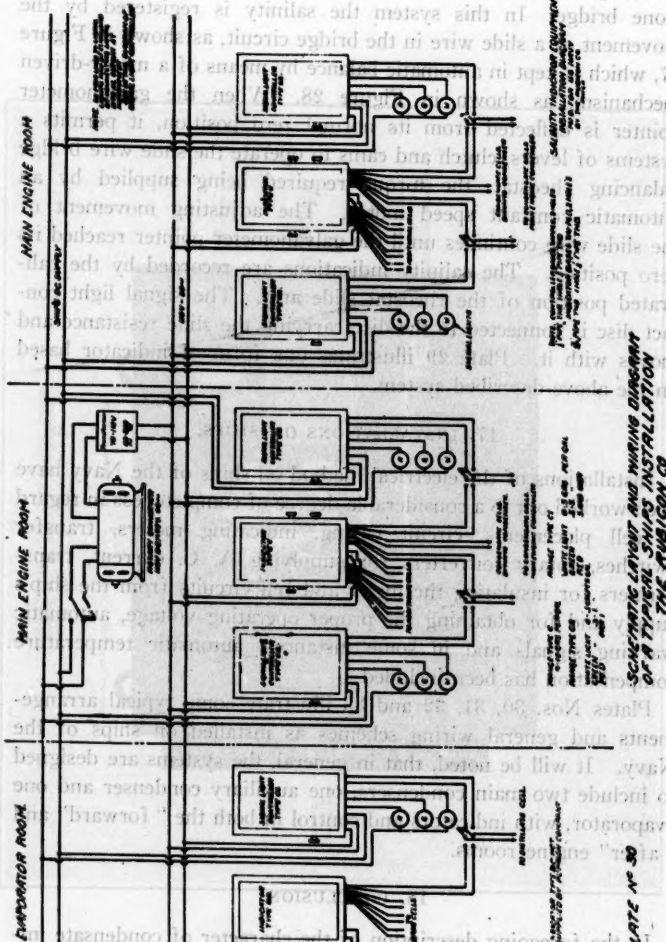
17. INSTALLATIONS ON SHIPS.

Installations of the electrical method on ships of the Navy have been worked out to a considerable degree of completeness in regard to cell placements, circuit wiring, indicating meters, transfer switches, rotary converters for supplying A. C. current, transformers for insulating the meter and cell circuits from the ship's supply and for obtaining the proper operating voltage, automatic warning signals and in some instances, automatic temperature compensation has been included.

Plates Nos. 30, 31, 32 and 33 illustrate some typical arrangements and general wiring schemes as installed on ships of the Navy. It will be noted, that in general, the systems are designed to include two main condensers, one auxiliary condenser and one evaporator, with indicators and control in both the " forward" and " after" engine rooms.

18. CONCLUSION.

In the foregoing description of the character of condensate impurities, the effect of marine impurities on boiler operations, and the method used to determine the amount of impurities in the condensates and feed waters, attempt has been made in a general way, but in sufficient detail, to enable the engineer to obtain an


MAIN ENGINE ROOM
MAIN ENGINE ROOM
EVAPORATOR ROOM

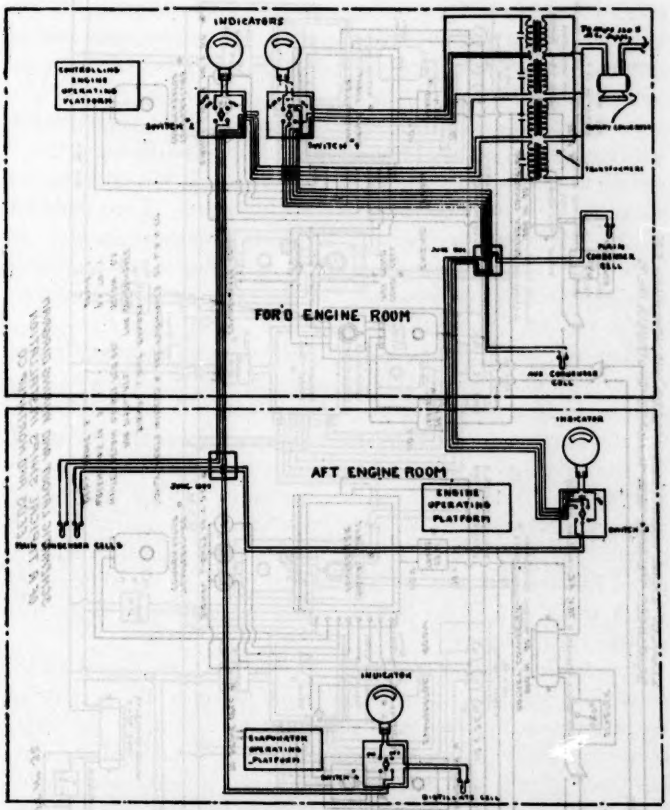
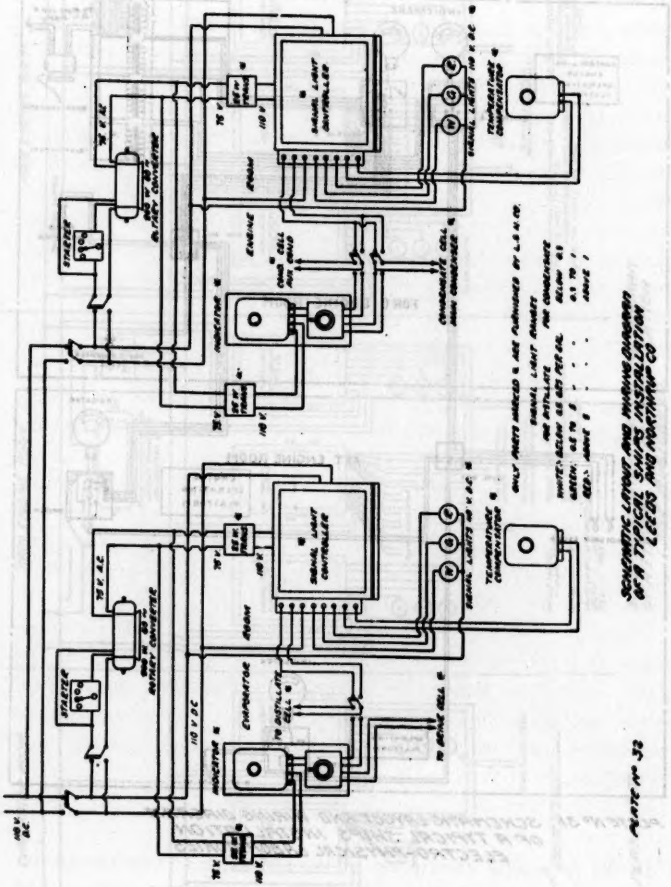


PLATE N° 31 SCHEMATIC LAYOUT AND WIRING DIAGRAM OF A TYPICAL SHIPS INSTALLATION ELECTRO-PHYSICAL LABORATORIES



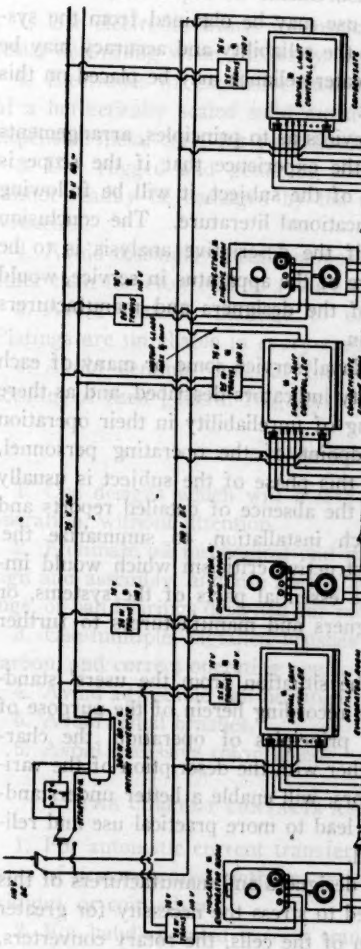


PLATE N° 33

SCHEMATIC LAYOUT AND WIRING DIAGRAM OF A TYPICAL SHIPS INSTALLATION. LEEDS AND NORTHRUP CO.

insight into the possibilities and limitations of the various systems of control now in use. Also by a study of the various principles involved in the electrical measurements of impurities, it is considered that greater practical use may be obtained from the systems now in service, and that the reliability and accuracy may be increased to a point where greater reliance may be placed on this important engineering control.

In describing engineering devices as to principles, arrangements and application, it has been the experience that if the scope is restricted to just such phases of the subject, it will be following along the lines of general educational literature. The conclusion of the article at this point, if the descriptive analysis is to be considered fairly representative of the apparatus in service, would possibly meet the approval of the designers and manufacturers of the systems described herein.

However, as there are in Naval service some or many of each of the types of electrical salinity indicators described, and as there appears to be a general feeling of unreliability in their operation on shipboard from the standpoint of the operating personnel, further information affecting this phase of the subject is usually in demand. It is difficult, in the absence of detailed reports and personal investigation of each installation, to summarize the unreliability and to offer constructive criticism which would improve the use and care of the essential parts of the systems, or which would enable the designers and manufacturers to further perfect their schemes.

For the improvement of the situation from the users' standpoint, it is considered that the recording herein of the purpose of the measurements, the basic principles of operation, the characteristic salinity curves, together with the description of the various types of cells and indicators, will enable a better understanding of the subject which may lead to more practical use and reliance on the indication obtained.

From the standpoint of the designers and manufacturers of this class of apparatus, it is desired to stress the necessity for greater ruggedness in the construction of the cells, the rotary converters, the current contact and transfer points, and the indicator mechanisms.

In detail, these suggestions are outlined briefly as follows:

THE CELL.

1. Use electrodes which are mechanically very rugged, so as to resist deforming when carelessly handled.

2. Electrodes of expensive base metal, should be in the form of a hermetically sealed solid metal cap, mounted on a solid inexpensive metal core, for ruggedness and least expense.

3. Use rugged and generous sized attachment fittings which cannot readily be damaged by the use of the heaviest make-up wrenches.

4. Avoid dissimilar metal compositions in the cell construction, other than the electrode, to reduce local electrolysis.

5. Avoid metallic platings to obtain base metal electrodes. Platings are unreliable in every respect.

6. Avoid ceramic material for insulators and separators, on account of their general fragility.

THE ROTARY CONVERTER.

1. Use designs which will insure unlimited continuously quiet operation, without attention.

2. Eliminate all mechanical end thrust of rotor, by correct design and assembly, and use ample, automatic, oil-fed sleeve bearings, or ball bearings designed for radial load only.

3. Use multiple, pigtailed brushes with correct Navy grade of carbon, and correct operating angle.

4. Avoid adjustable ball bearings.

5. Avoid radial brushes.

6. Avoid excessive rotor speeds.

THE CURRENT CONTACTS AND TRANSFER POINTS.

1. For automatic current transfer points, use self-cleaning contacts of generous area, with arc-resisting metal, such as platinum, iridium, or coin silver.

2. For hand-operated current transfer points, use self-cleaning contacts of rugged design, and non-oxidizing surfaces, such as tin or nickel.

3. Use solderless mechanical terminals on wire connections to terminal posts.

4. Use rustless spring lock-washers in securing all electrical and mechanical joints.

5. Avoid the use of solder and fluxes in securing wires and terminal connections.

6. Avoid the use of exposed, unplated, iron, brass, or copper parts, where resulting rust or verdigris from salt air may affect the operation.

THE INDICATING MECHANISM.

1. Galvanometers should have high torque ratio, be well damped, have large air-gap clearances, and have rugged easily adjusted suspensions.

2. Mechanisms should be contained in a moisture-proof housing, and should give long continuous service without oiling or cleaning.

3. Incorporate a check test, fixed resistance coil of zero temperature coefficient adjusted to indicate the equivalent of one grain salinity at some standard working temperature, so as to check the instrument operation at this point.

4. Avoid high speed rotating parts.

5. Avoid the use of very fine wire windings for field coils and resistance units.

6. Avoid commutators and brushes.

The operating readers of the JOURNAL, who may be divided into two classes, viz., those who care to know how the systems operate, and those who desire to know why they fail to operate, for the latter class, by reading between the lines of the foregoing suggestions, one may possibly find the causes of service failures experienced on shipboard and perhaps apply remedial measures to increase the reliability of the installed systems.

RUBBERIZATION AS A PROTECTION AGAINST CORROSION AND ABRASION.

By Roswell B. Daggett, Lieutenant (CC) U. S. Navy, Member.

In the Journal of the American Society of Naval Engineers. Volume XXXVI No. 2 of May 1924, Commander P. E. Dampman, U. S. Navy, discussed rubber sleeves for the protection of steel shafting from corrosion. Since that time there have been improvements in the method of application of rubber to shafting, and the Navy Yard, Boston, has extended the use of rubber to propeller shaft struts. It is the purpose of this article to discuss the recent developments in rubberization of parts of ships constantly subjected to the action of salt water.

The chief improvements in applying rubber sleeves to shafting have been in the type of joint used where the rubber makes up on the composition sleeve and in the use of aluminum rubber. When the *Denver* class of cruisers were dry-docked after having used their first rubber sleeves for nearly a year, it was found that water was reaching the steel shafts by passing under the rubber at the joints next to the composition sleeves. This was evidenced by rust appearing at the surface of the rubber. The rubber was removed and the shaft found to be slightly pitted near the joints; otherwise the rubber had provided adequate protection for the shafting throughout the length of the sleeve.

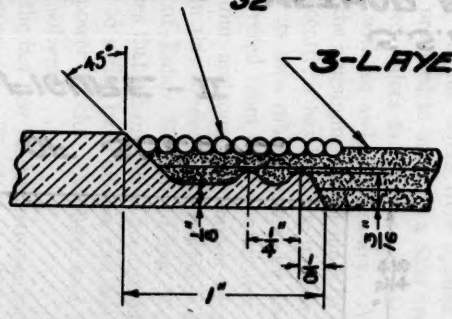
The original joints were somewhat similar to those found on the S. S. *Leviathan* shafts as installed by the Germans. Each section of composition sleeve was underscored at the ends by a 45 degree bevel from the outside diameter of sleeve to shaft. The rubber was then tucked into these recesses and vulcanized on to the shaft by the use of carbon bisulphide. It is true that rubber expands when wet with carbon bisulphide, but later, as the carbon bisulphide evaporates, the rubber shrinks. This meant that be-

sides the mechanical difficulty of properly working the rubber into the 45 degree recess, the shrinkage of the rubber caused it to leave the metal walls of the recess and provided an access for the water to reach the shaft. The reason why this had not occurred on the *Leviathan* shafts was due to the different method of application used by the Germans. It is thought they applied the rubber to the shaft under pressure and then placed the shaft in a steam vulcanizer whereby the rubber was vulcanized into the recess by heat.

Up to this time, it was not possible to make rubber vulcanize to composition. However, the fact that rubber shrinks upon the evaporation of carbon bisulphide was taken into consideration in the new design of joint. The outside of the composition sleeve was turned down in a lathe, as shown in Figure I. The rubber was then carried up over these steps, care being taken that the diameter of the rubber sleeve remained slightly less than the diameter of the composition sleeve. The shrinkage of the rubber during the evaporation of the carbon bisulphide then made the rubber grip the composition. But since it was known that the rubber would not adhere to composition, a joint was made by wire bands wound tightly around the rubber over the steps. Other than this modification in the joint, the rubber was applied as before, and at the next dry-docking no evidence of leakage was noted. The rubber was not removed from the shaft for inspection as it was thought to be providing perfect protection.

Recently, however, the United States Rubber Company has developed an aluminum rubber, No. 2864F, which will adhere to composition as well as to steel and which is now being used in the location of the joints, eliminating the necessity of wire bands. This was used in providing rubber sleeves for a new tail shaft for the S. S. *Leviathan* just completed by the Boston Navy Yard. The composition sleeve at the joints was prepared as shown in Figure II. The entire area to be rubberized was washed with carbon bisulphide to remove all foreign matter. Two coats of aluminum cement as prepared by the Revere Plant, United States Rubber Company, were applied. When these became tacky to touch, carbon bisulphide was applied freely, and then a sheet of aluminum rubber ⅛ inch thick was wrapped around the shaft

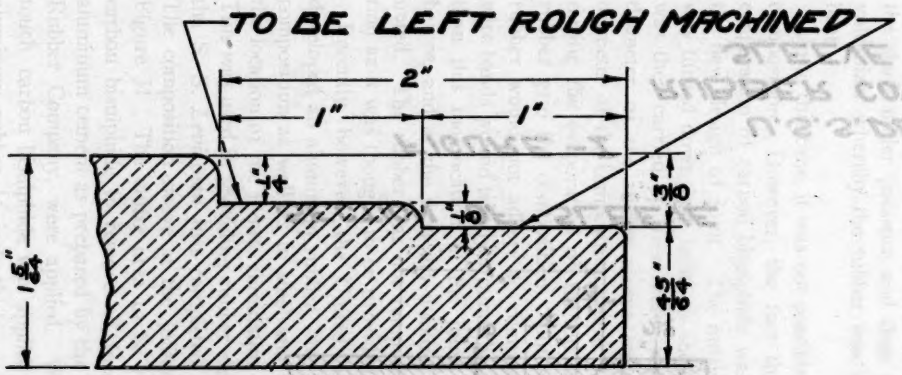
ENDS OF RUBBER COVERING TO BE TIGHTLY WOUND WITH BRONZE WIRE - $\frac{3}{32}$" DIA.



SECTION OF SLEEVE

FIGURE - I

U.S.S. DENVER
RUBBER COVERING AT
SLEEVE ENDS



SECTION OF SLEEVE

FIGURE - II

S.S. LEVIATHAN
METHOD FOR MACHINING
SLEEVE ENDS FOR
RUBBERIZING SHAFTS

making up in a scarfed longitudinal joint. Carbon bisulphide next was applied generously and the sleeve built up to the required thickness with sheets of rubber ⅛ inch thick, known by the United States Rubber Company as 2865F. Before the last layer of rubber was fitted, rubber cement followed by a layer of breaker strip (which is a rubber with inserted cords) was applied for the purpose of giving mechanical strength to the rubber sleeve. The last layer was painted freely with carbon bisulphide and the entire sleeve wrapped with cotton stripping for twenty-four hours at about 60 degrees Fahrenheit. This completed the curing and made a homogeneous sleeve in which the several layers had become one. As the layers were applied, additional strips at the end were provided to build up a fillet and each layer was worked with knurled rollers to eliminate all air bubbles, the sheet being punctured with an awl as necessary. All applications of rubber following the aluminum cement must be accomplished in one day.

Until recently, it has been the policy of the Boston Navy Yard to prepare its own rubber cement by dissolving the uncured rubber as delivered from the manufacturer in benzol in the proportions of four and one-half pounds of rubber to one gallon of benzol. In this connection, attention is invited to the U. S. Department of Labor Bulletin of the Women's Bureau, No. 57, which states that benzol fumes are extremely poisonous and cause the victim to suffer from anemia and hemorrhages since the blood loses its power to clot. Naphtha and benzine are much safer petroleum solvents.

During the last few years, destroyer propeller struts have shown excessive corrosion. The Bureau of Construction and Repair of the Navy is much concerned over this condition which is constantly becoming worse. Fears are not entertained for the strength of the struts as they were originally designed with a large factor of safety, but no method of arresting this corrosion has been proven adequate. Underwater zincs have been re-installed around the stern; the spots showing greatest corrosion have been filled with welding metal; iron cement (smooth on) has been applied to the struts; and at the next dry-docking, increased corrosion would be in evidence. Recently, certain navy yards have provided the struts with wrapper plates after which red lead putty

has been pumped between the wrappers and the struts. The efficiency of this method, however, for keeping the salt water away from the struts has not yet been determined.

The destroyer *Putnam* was dry-docked in the Navy Yard, Boston, on April 11, 1927. Her struts showed very excessive corrosion, particularly on the top surface of the horizontal arms and where these arms joined the boss. There were certain spots where a pocket-knife blade could be inserted to a depth of one inch. The possibility of rubberizing these struts was suggested, since this method had proven so satisfactory in shaft work. A representative of the Revere Plant, United States Rubber Company, inspected the struts and stated that it would be very possible to provide efficient rubber sleeves. Authority was requested and received from the Bureau of Construction and Repair to proceed with the work, and it was decided to rubberize only the two horizontal struts, which were in the worst condition, as an experiment. If this installation proves successful, the rubberization may be extended to take in the upper struts and the bosses.

The corrosion, which was general, appeared at dry-docking as shown in Figures III and IV. Although the underside of the strut was not in as bad condition as the upper, it was decided to envelop the strut entirely with rubber. The metal surface was thoroughly sandblasted to remove all rust, paint, and foreign matter from the area to be rubberized. Great care was taken to clean thoroughly to bare metal the bottoms of all pits. Norwesco iron cement ("smooth-on" as carried in stock in the Navy Yard) was used to fill in the deep cavities flush with the surface of the strut, but none was applied to the surface itself. This was done to prevent the formation of an air pocket in those holes too deep and irregular for the sheet rubber to reach. At least twenty-four hours must be allowed for the smooth-on to set before proceeding with the rubberization.

The struts were then washed thoroughly with naphtha to remove all grease and oil. A light coat of rubber cement, U. S. Rubber Co. No. 2853F, was applied followed by as heavy a coat as possible of the same. As soon as this became tacky to touch, two heavy coats of rubber cement, U. S. Rubber Co. No. 2863F, were applied as smoothly as possible with a brush. The cement

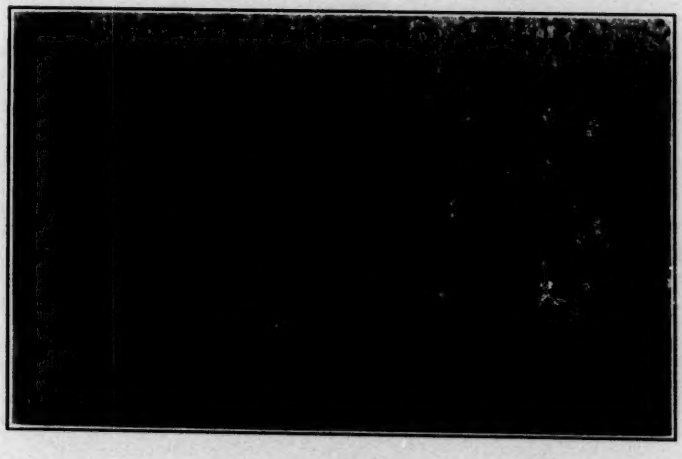


FIGURE III.

FIGURE IV.

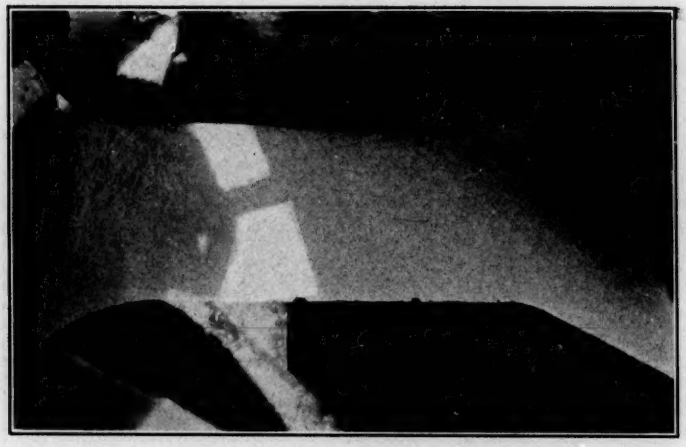


FIGURE V.

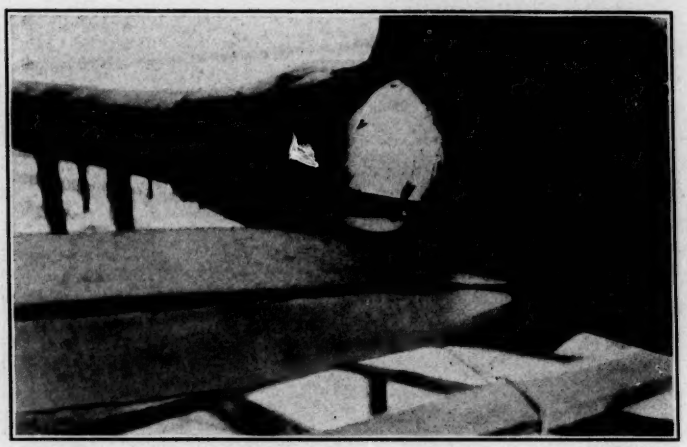


FIGURE VI.

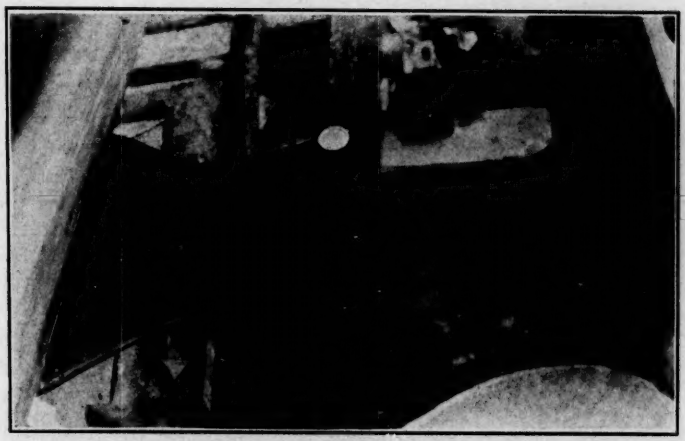


FIGURE VII.

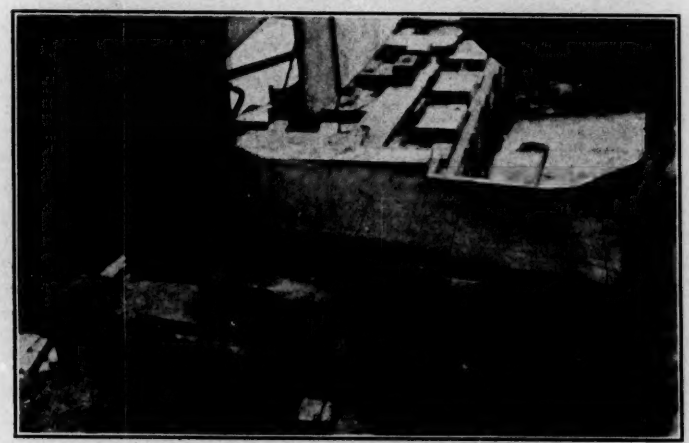


FIGURE VIII.

FIGURE IX.

was given eight hours to set at 70 degrees Fahrenheit. The appearance of the struts after the cement was applied was as shown in Figure V.

Immediately previous to applying the uncured rubber, the cement was washed with naphtha. The rubber was 5/32 inch thick, U. S. Rubber Co. No. 2867F, and was put on in two sections. The top surface of the strut was first covered by a sheet cut to size and allowed to overlap the trailing and leading edges of the strut by about five inches. A similar layer was provided for the bottom surface and overlapped on to the top layer about five inches. Seams were scarfed. This stage of the operation is as shown in Figure VI. As each layer was applied, it was worked with small knurled rollers to eliminate all air pockets, the rubber being punctured with an awl as necessary. The rubber is provided in rolls with cotton sheeting on one surface of each roll to prevent it from forming a mass. It was unrolled and cut to required dimensions with the sheeting removed. Both surfaces of the rubber were washed thoroughly with naphtha and a piece of sheeting placed under the rubber as the latter was placed in position on the strut. This prevents the rubber from adhering to the strut before properly fitted. The cotton was then removed with care not to change the position of the rubber relative to the strut. The lapped surfaces of rubber were wiped with naphtha just previous to lapping. Carbon bisulphide was applied freely in three coats to chemically cure the rubber and make into one layer the two layers where lapped. This condition is shown in Figure VII.

As soon as the rubber felt sticky after the carbon bisulphide had been applied, the strut was wound with one layer of stripping tape, U. S. Rubber Co. No. 2867F. This is an eleven-ounce duck impregnated with rubber and is provided in strips six inches in width. The tape was wrapped tightly around the strut in single strips parallel with the keel of the destroyer. All edges were butted, as shown in Figure VIII, after which carbon bisulphide was applied freely. This was allowed to cure chemically for thirty-six hours at about 50 degrees Fahrenheit. The length of time required for curing depends upon the temperature. At 80 degrees Fahrenheit, twelve hours would be required; and at 212 degrees Fahrenheit but two hours. An excellent test to determine if the

rubber is properly cured is to test its hardness. When thoroughly cured, the rubber shows great resistance to indentation. In this case, however, certainty of the vulcanization was made by wrapping the struts loosely with canvas, as shown in Figure IX, and supplying steam from an open ended steam hose under the canvas at about 15 pounds pressure for four hours. An inspection showed the rubber to be adhering to the steel at the edges in an excellent manner, and the *Putnam* was undocked within three hours.

The Boston Yard does not propose to rubberize any more struts until the *Putnam's* installation has been inspected at her next dry-docking. If no rust appears at the edges of the rubber at that time, there will be no need of removing the rubber for examination, as the exclusion of salt water from the struts will undoubtedly have been obtained. There is no apparent reason why this method will not prove as successful for struts as it has for tail shafts. Both the struts and the tail shafts are exposed to approximately the same flow of water at the run. Also, the increased thickness of the struts due to the rubber is not thought to add seriously to the resistance to propulsion.

The struts of the *Putnam* were prepared by employees of the Navy Yard, Boston, up to the point of applying the rubber cement. From then on to completion, the rubberization was done by employees of the United States Rubber Company experienced in handling rubber. This procedure was followed when Boston applied rubber for the first time to the shafts of the *Denver* class. However, the Navy Yard employees now experienced in shaft rubberization assisted in the strut rubberization, and it is the intention of the Yard to have these employees do all strut work which may be done in the future. By this plan, rubberization would not be an expensive process, and much time would be saved in having Yard personnel experienced in this work.

There are certain features in the fitting of the rubber which it is thought can be improved upon at a future strut rubberization. The layer on the bottom half will be applied first and lapped about four inches over both strut edges and cemented to the top half. The edges of the lap will then be scarfed and a layer of rubber cemented on top of the strut to make up on these scarfs. This will eliminate a double thickness for a distance of four inches on

the top and bottom surfaces next to the strut edges. Also, it is thought that it would have been better to have overlapped the edges of the stripping tape about ¼ inch. A better application of the tape could have been made if it had been cut on a bias.

In addition to shafts and struts, the United States Rubber Company has extended the use of rubber in marine work to pump rotors and cast iron propellers. The life of rotary scoops handling abrasive material is greatly prolonged by rubberizing the metal parts. The abrasive particles impinge on the rubber and rebound without wearing away the rubber. Cast-iron propellers are painted with a dissolved cured rubber which forms a brittle rubber coating as soon as the solvent evaporates. This application, due to its brittleness, will flake off if the propeller receives mechanical injury. Experimentation may produce a rubber solution which can be applied by painting or dipping and which will have resilience after the evaporation of the solvent. There are places, however, where brittleness would not impair the use of this rubber; such as, battery compartments and submarine battery ventilation piping and fans which are not likely to receive mechanical injury. At present, these installations require ferquent renewal due to the action of the battery fumes. An appreciable saving in weight could be effected by substituting rubber tiling for ceramic tiling. Those parts of ash-ejector piping on coal-burning ships which require frequent renewal would undoubtedly have greater longevity if rubberized. The above suggestions indicate a few possibilities for the application of rubber in marine work as a protection against corrosion and abrasion.

NOTE: A recent development by the United States Rubber Company has been the rubber covering of struts, employing the same method as used in the rubber covering of propeller and stern tube shafting wherein metal bands are used in conjunction with the rubber as an additional locking medium, thereby entirely eliminating the use of frictional duck or tape.

MODIFIED BACK PRESSURE VALVE.

By C. J. Odend'hal, Lieutenant Commander (E) U. S. Coast Guard, Member.

One of the requirements for obtaining engineering efficiency and economy is to maintain a steady back pressure. With the back pressure valves usually installed in vessels someone has to be constantly on the alert opening and closing the valve to hold the pressure in the exhaust line constant. As a rule, a man on watch will be very careful in this respect as long as there is someone in authority about, but will be careless otherwise. Unless a close regulation of the exhaust valve is kept, when additional machinery is started the exhaust pressure will rise. This results in the slowing down of all other machinery previously in operation. Stopping machinery has the opposite effect of reducing the back pressure with the consequent speeding up of auxiliaries. This is particularly noted when preparations are made for getting under way for it is then that the variation in the number of machines in use is greatest. It is then, too, when the services of all men in the department are at a premium, yet the service of one man is practically wasted attending a valve that should be self regulatory. The rise in pressure in the exhaust belt with increased number of auxiliaries in use is due to the fact that the spring tension tending to close the back pressure valve increases as the valve opens. The reverse should be the case. An increase in volume of steam in the exhaust will cause the back pressure valve to lift. As soon as the valve opens, the pressure under the valve will drop, and to balance this the tension on the spring should be reduced instead of increased. This condition is aggravated the further the valve opens until there is a point where the tension on the spring has to be eased off materially to allow the valve to lift a sufficient amount. A study of the drawing will show an attempt to approximate this.

Figure 1 is a drawing of the entire valve showing the horizontal links pivoted to the valve stem at one end, with the roller at the

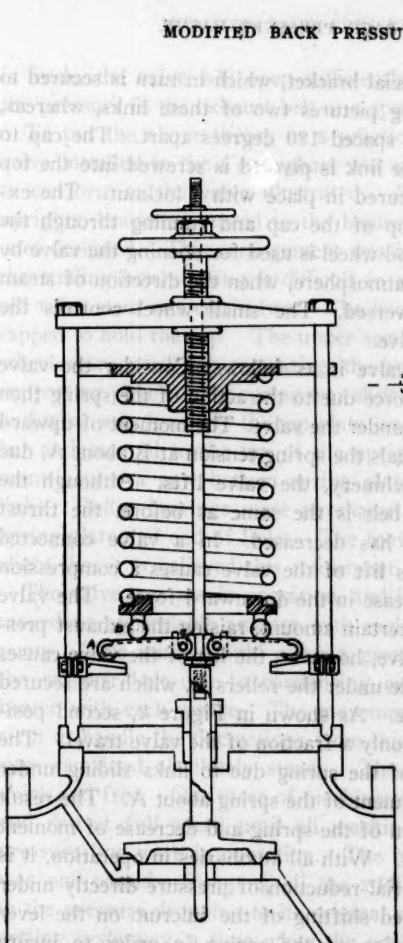


Fig 1

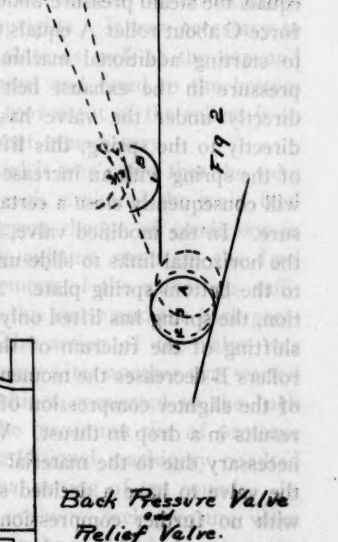


Fig 2

Back Pressure Valve and Relief Valve.

other end sliding on a special bracket, which in turn is secured to the upright. The drawing pictures two of these links, whereas, in reality there are three, spaced 120 degrees apart. The cap to which the inner end of the link is pivoted is screwed into the top of the valve stem and secured in place with a locknut. The extension attached to the top of the cap and running through the pressure adjusting stem and wheel is used for opening the valve by hand when exhausting to atmosphere, when the direction of steam through the valve is reversed. The small wheel controls the manual opening of the valve.

The operation of the valve is as follows: Consider the valve closed. The downward force due to the action of the spring then equals the steam pressure under the valve. The moment of upward force C about roller A equals the spring tension at B about A, due to starting additional machinery, the valve lifts. Although the pressure in the exhaust belt is the same as before, the thrust directly under the valve has decreased. In a valve connected directly to the spring, this lift of the valve causes a compression of the spring with an increase in the downward force. The valve will consequently close a certain amount, raising the exhaust pressure. In the modified valve, however, the lift of the valve causes the horizontal links to slide under the rollers B, which are secured to the bottom spring plate. As shown in Figure 2, second position, the spring has lifted only a fraction of the valve travel. The shifting of the fulcrum of the spring due to links sliding under rollers B decreases the moment of the spring about A. The result of the slighter compression of the spring and decrease of moment results in a drop in thrust. With all auxiliaries in operation, it is necessary due to the material reduction of pressure directly under the valve to have a decided shifting of the fulcrum on the lever with no further compression of the spring in order to insure proper opening of the valve. Figure 2 illustrates the tendency of the roller A to shift more rapidly to the right, and roller B to lift the spring less and less as pivot C rises. A further rise of C will cause no further compression of the spring with a decided shifting of roller A to the right. Accurate regulation can be obtained by the proper position of roller B on link A, and correct slope of the platform of the bracket secured to upright. After this adjustment

is made, the valve will open sufficiently at all times to maintain the pressure in the exhaust belt at a specified setting.

To put the above theory to a practical test, the back pressure valve installed in the *Conyngham* was modified as far as possible to conform to the valve illustrated. As the valve had but two uprights, brackets secured to the bonnet were substituted for the ones designed. This arrangement was not desirable as it crowded the stuffing box, making it difficult to set up on the gland. The old valve stem was cut in two. The lower section was drilled and tapped to hold the cap. The upper section of the valve stem with wheel were used for regulating the spring setting. The lower supporting plate for the spring was shifted to the top of the spring, and what was formerly the top plate was used as the thrust member and secured to the top of the uprights. It was necessary to manufacture a new bottom spring plate as well as the cap and links. Roller bearings were purchased and fitted to the bottom spring plate and end of links. The movement of the rollers is so slight that it is believed this refinement unnecessary.

The valve was put in operation, and it was noted that the stem moved an appreciable amount with each stroke of the main feed pump. This pump would cause a great change of conditions in the exhaust belt due to condensing exhaust steam in the feed heater with each stroke. The movement of the valve was sufficient to handle the fluctuations from this cause and the back pressure remained absolutely steady. The action of the valve was quick and free. One piece of machinery after another was started and run at full speed until all machines connected to the back pressure line were in operation. The maximum rise of pressure was one pound. Stopping all the additional machinery resulted in the pressure dropping to its original setting of ten pounds. In getting underway, a time when the ordinary back pressure valve acts most erractically, no one went near the modified valve to make any adjustments. The maximum rise of pressure was four pounds. This rise in pressure occurred only when maneuvering and was due to a change in vacuum. Underway the valve performed satisfactorily. Whereas before the back pressure varied from five to twenty pounds, requiring almost constant regulation, with the new valve an adjustment was necessary only at change

of speeds. This was required as the pressure on top of the valve changed with the pressure of the stage into which the exhaust steam was turned. Once adjusted, however, for that speed, no further regulation was required. As the gland steam is taken from the exhaust line, variation in the exhaust pressure formerly caused the glands at one time to leak an excessive amount of steam, and at another to allow the air to enter the turbine with consequent loss of vacuum. No trouble of this kind has been experienced since the installation of the modified valve in question.

It is believed that a valve similar to the one mentioned could be used to advantage in the fuel oil burner line to iron out the fluctuations of the pump, allowing the excess oil to return to the suction side of the pump.

NOTES.

NATIONAL DEFENSE AN INVESTMENT.

By Dwight F. Davis, Secretary of War.

(An Approved Interview with Theodore M. Knappen.)

(Because of specious propaganda that national expenditures on the Army and Navy are pure waste, "The Magazine of Wall Street" asked Secretary Davis to present national defense in its true light as a fundamental investment.—Editor.)

The pacifists are wont to assert that 86 per cent of all our national expenditures go for wars, past, present and future, and they hope that Congress will reduce that proportion through reducing present military expenditures. The logic of this attack is somewhat indirect, but the theory is that expenditures for wars and expenditures for maintaining an army and navy are of one piece and equally useless and wasteful. The argument may be a bit mixed but it has the force of an appeal to the pocketbook.

"Is it true," I asked Secretary Davis, "that 86 per cent or thereabouts of our national expenditures are chargeable to war account?"

"Before dealing with the true nature of war costs and national defense appropriations," the Secretary replied, "it should be noted that the 86 per cent is, to say the least, somewhat exaggerated. It is obtained by charging the whole service of the public debt to past wars and adding thereto current military appropriations and expenditures for pensions and allied items.

According to the Budget classification, the military functions of the United States Government actually are about 30 per cent of the cost of the Federal Government, and that includes pensions, compensation, war insurance, and the like. Our current War Department expenditures are only about 13 per cent of the whole governmental outgo—and a large part of them are of a non-military nature.

As to the implication that the Army and Navy are in any way chargeable with the past wars that our pacifist friends wish to load them up with, I would ask whether there is any reasonable man or woman in the United States who really believes that our military organizations ever brought upon us a single war? It may be, and doubtless has been, different with other nations, but if there is one certain fact in our 150 years of national life it is that our armed forces have never been political or influential in determining the issue of peace or war.

Our forces have been no more responsible for our wars than our police officers are accountable for crime. Our soldiers and sailors have merely fought our wars on orders from the Government."

"But if the Army and Navy are to be charged with wars which they fought to end, instead of bringing on," I remarked, "then they should be credited with the results of those wars, and also of all the other beneficial results of military organization and activity."

"If we can get the credit for the benefits of that fictitious 86 per cent,' was the reply, "we are willing to let it stand unquestioned."

In general that potential credit may be described as the whole of the wealth and greatness of the United States. It is what this nation is. It is the greatest investment in the world. The Secretary sketched it as follows:

"To begin with, there would have been no occupation of this continent if the settlers and pioneers had not always been soldiers and wagers of war on occasion. The title deed from the Indians to the whites, of the whole continent, was written with the sword. Next, it was military force that decided whether the culture of the continent should be British or French. As our ancestors were part and parcel of the English dominion and culture, we may assume that British victory was of supreme importance.

When we come to the War of Independence, the fighting men may justly claim credit for its results if they are to be debited for its costs. The Revolution cost about $50,000,000. Was it a good investment? Or was it just a waste of $50,000,000 which the poor taxpayers had to bear when they might have had tyranny and colonial subjection without effort—and their $50,000,000 to spend for rum and necessaries? To be sure the Revolution bill has been paid, and so does not appear in the 86 per cent, but the principle is the same.

The War of 1812 was a very messy sort of conflict. It was certainly not brought on by our professional fighting men, and they did not have much to do with directing it after it started. But somehow it was fought and substantially, if not gloriously, won; and the soldiers, whether on land or sea, won it. But if it is to be blamed on the fighting men, then they may justly urge that as against the $200,000,000 or $300,000,000 it cost, there was the supreme asset of national independence maintained, of freedom of the seas achieved, of the northern borders definitely asserted, of American commercial rights defended; and of the imperial Louisiana Purchase nailed down, as against British desire to wrest that domain from us. What was the cost of the War of 1812 against the potential and since largely realized wealth of that vast sweep of country from the Gulf of Mexico to the Rocky Mountains and the 49th parallel?

The War with Mexico may or may not have been morally creditable, that is not for me to say; but it was a very material success. It gave us New Mexico, Arizona, California, Nevada, Utah, Colorado, and assured Texas. Was that territorial expansion worth the trifling cost of the war? Who could find the debit of the Mexican war in the nation's credit account with the Army and Navy?

As with its predecessors, the Civil War was not a product of readiness for war; rather of unreadiness. The late General Leonard Wood once said that if the nation had had a real army of 50,000 men the domestic quarrel would not have lasted ninety days. However, it took soldiers to terminate a war they did not cause and were not permitted to prevent. Was it worth three billion dollars to maintain the unity of the Republic?

Whatever was gained by the war with Spain—Porto Rico, the Philippines, Guam—is to the credit of the men who settled with the sword that which proved impossible to compose in the council chamber. It must always be remembered that when the civil authorities declare a state of war the event is in the hands of the military.

As with all our wars, we entered the World War unprepared. The Army, always true to Washington's dictum that preparedness is the best assurance of peace, had done what it was permissible for it to do in the way of urging preparation, but little was done. After the struggle began the very people who opposed preparedness condemned the Army for its unreadiness! Almost

a year elapsed before we had a respectable force in France and first to last we had to depend upon the French for virtually our entire artillery equipment. This war cost us $34,000,000,000. That was not the Army's responsibility, but the responsibility of the whole people. The Army's task was to bring the war to an early and successful conclusion. Who can appraise what a material loss defeat would have meant? Being in the war, victory was well worth the $34,000,000,000.

The terrific pace at which we had to go to make up for preparedness resulted in the waste that usually accompanies haste. It may be truly said that much, if not the larger portion, of our stupendous World War bill, was not the cost of war waging, but of war raging. I say this in no criticism of anyone except it be the nation as a whole. It is our fatuous national habit to affect a bland confidence in unbroken peace, and yet get into a serious war every generation; and, of course, always unready.

It is quite possible that if we had spent a billion dollars getting our Army ready for eventualities, after the world caught fire in 1914, we would never have been drawn into the World War. It was German knowledge of our unreadiness that led the war lords to take the risk of tempting us to join the allies. And the justification of the risk is that it almost won. I mention these things to show how unreasonable it is to couple the Army and Navy with financial responsibility for our wars. To do so is like blaming the fire department for the outbreak of a fire, the doctor for the inception of the patient's illness, the repair crew for the railway wreck. The fireman, doctor and the repairmen have to be paid, but they are not to be charged with the accident or fault that made their services necessary. All too little have we used our military forces as preventive, all too much as surgical, treatment."

People recall only the foreign wars and the Civil War. The incessant Indian wars have almost faded from their memories. There was not a year and scarcely a month from the Indian battles of St. Clair and Wayne in Ohio in 1792-94 until the Apaches were crushed in 1886, that our little Regular Army was not in some skirmish or battle with the Indians. The occupation of the whole region from the Alleghenies to the Pacific was behind a thin line of heroic Regulars who had many a bloody task that they abhorred. Fighting the American Indians was no child's play. They were born soldiers of high natural intelligence; they developed war chiefs of exceptional military ability; and, given equal equipment, they were, man for man, a match for our soldiers.

"If we assume that military force was indispensable to the conquest of the west—and that is a reasonable assumption"—the Secretary commented, "then must we credit the Army with the occupation and pacification of a region that has become the granary of the nation, the scene of prodigious industry and the home of a majority of our people. Was the cost of the Indian wars a good investment?"

"What of the cost and investment value of the Army and Navy in time of peace," I asked.

"Is $600,000,000 a year too high for some measure of preparedness, of insurance for a national wealth that approaches $400,000,000,000? The purpose of our military organizations is to prevent wars as well as to fight them when they come. We all hope and pray for perpetual peace, and no one more sincerely than the man who will have to do the fighting, but it seems to me childish to imagine that we would have peace and our present security if we were disarmed."

"What about the non-military tasks of the Army and Navy?"

"To their credit as victor in all our wars there is to be added a great score of essentially civil, directly productive and humanitarian, work. About one-

third of the annual cost of the Army is for non-military functions. From the days of Lewis and Clarke to this age of aviation the Army has been the nation's pathfinder and explorer, engineer, geologist, road builder, surveyor, investigator in medicine and surgery, rescuer in fire and flood, repressor of civil disorder and the ready means of meeting a variety of emergencies.

Almost every railroad built in this country before 1855 was located, constructed and managed by army engineers. Among them were the Baltimore & Ohio, the Northern Central, the Erie, the Boston & Providence, the New York, New Haven & Hartford, and the Boston & Albany. Practically all the continental lines were projected by the Army. It built the Erie and Chesapeake & Ohio canals and the great national highway from Cumberland, Md., to St. Louis. The boundaries of the United States and most of the state boundaries were surveyed and marked by the Army. Most of our lighthouses were raised by the Army. Our harbors and sea channels and our inland waterways everywhere bear the impress of the Army. The Army completed and the War Department is associated with the administration of the Panama Canal. The foundations of government were laid by the Army in Alaska, Hawaii, Cuba, Porto Rico, and the Philippines.

The Signal Corps of the Army laid the pioneer telegraph lines throughout the west. The Army has led in the development of aviation and the radio. It even founded our great steel industry through its requirements and specifications.

The practice of making standard interchangeable parts, the very essence of mass production, was originated by the American Army in the manufacture of rifles as long ago as 1798. The greater part of the engineering work of the Federal Power Commission in administering the water-power resources of the Government falls to the Army engineers. The weather bureau began in the Army. The army doctors conquered yellow fever and learned how the white man might master the heat and diseases of the Tropics.

Our Army and Navy have ever been great schools for success in civil life. During the first century of its existence West Point alone contributed 2,371 men to eminent positions in civil life—men who took a great part in creating our national patrimony. The ranks of the Army are a vast trade school, which annually sends thousands of trained and disciplined men back into civil life. The disciplined army veteran is the preferred policeman and state trooper, and an honorable discharge from the Army is the best introduction a man can have to a job.

Taking it altogether the Army and Navy can well stand the debit of all the billions and billions our wars and pensions, compensation and insurance have cost us, if they must be charged to men who did not incur them, as against the credit that must likewise go to them for the unbroken record of protection of the largest and best paying investment in the world—the United States of America."

Following up Secretary Davis' conception of the United States as the most extensive and most lucrative investment ever made; and to be credited to our military forces, if the cost of all our wars is to be charged to them, here is the way the balance sheet stands.

Debit

Cost of wars, $40,000,000,000

Credit

Wealth of the United States, $400,000,000,000

Viewed as a going concern, this colossal investment, secured by the Army and Navy and defended by them against a covetous world, pays dividends at the rate of $90,000,000,000 a year—the national income.

Secretary Davis is right, never was there such an investment and, let us add, never one so well worth defending.—"The Magazine of Wall Street."

PULVERIZED FUEL FOR MARINE PURPOSES.*

By Engineer-Captain J. C. Brand, R.A.N.

The term powdered fuel as used in this paper is meant to include any carbonaceous fuel which is in a sufficiently divided state to be impelled by or borne in air, and burned while in a state of suspension. Since the fuel to be ground is reduced in size by mechanical effort, it is more economical to grind slack or breeze than larger pieces. Not only is this remunerative mechanically, but also financially. Small fuel is of less value on the market than larger sizes, which possess a more extended market. Fuel containing up to 30 per cent of ash may be burned, but it must be borne in mind that the ash is unproductive of heat and must be ground, conveyed, and disposed of. Including all charges, it may be assumed that the cost of pulverization is about 2s. 8d. per ton on an output of 80 tons per 24 hours. Therefore, compared with fuel of 7 per cent ash, 9½d. has been expended uselessly in the case of fuel with 30 per cent of ash, contrasted with 2¼d. in the case of the 7 per cent of ash. In the case of dry fuel of 13,000 B.T.U.'s per pound having 7 per cent of ash and 5 per cent moisture, to crush, convey, pulverize, blow, and feed it to the furnaces would absorb power equal to 2.2 per cent of its steam-producing capacity. In the case previously mentioned, *i.e.*, 30 per cent ash and 9,500 B.T.U.'s per pound and 5 per cent moisture, an equivalent of 3.0 per cent of its evaporative capacity would be absorbed. It is evident, therefore, that the fuel containing the greater percentage of ash needs careful financial consideration before it is accepted as cheaper than a superior one.

Combustion space in a marine boiler, whether of the Scotch or water-tube type, is limited, as is also the length of path available in which combustion must be completed. In consequence, long flame burners of the tubular or fish-tail type—commonly used in land practice—which were tried were abandoned in 1917 for use in the more strenuous conditions for which a solution was sought. Such a burner must be capable of effecting a complete combustion in a distance of about 6 feet; and since it is necessary for the particle to remain in the combustion zone for a certain period, it follows that to traverse a path of 30 feet, such as that often given in land boilers, the particles must follow a spiral path. The spiral will gradually straighten out as the influence of the energizer, *i.e.*, the primary air, dies out. The velocity of projection is about 70 feet per second, and at completion of combustion 40 feet per second, in the system with which the author is connected. The latter point is reached just before the mouth of the furnace proper.

Tests with the Brand burner, designed by the author, using air heated to 150 degrees C. and with coal containing 18 per cent of volatile hydrocarbons, gave a temperature of 1,480 degrees C. 15 inches from the burner mouth, combustion being complete at 5 feet. The kinetic energy imparted to a

*Abstract of paper read at the Summer Meetings of the Institution of Naval Architects held at Cambridge, 12th July, 1927.

small particle is less than to a larger one; but the total surface area of the smaller particles is infinitely greater than the same weight of larger ones. Since the kinetic energy of the lighter particle will be exhausted more rapidly than that of the larger one, it will remain longer in the intensely hot zone, surrounded by refractory material, in front of the burner. Bourgand ("Metallurgical and Chemical Engineering," Vol. 24, 1921) shows that the higher the temperature the more rapidly the reactions are completed. At 1,350 degrees C. the reactions are about four times as rapid as those at 1,220 degrees C.

Moisture in fuel, inherent and adherent, has a detrimental effect on freedom of flow, without which certainty and evenness of operation are impossible. This applies equally to bunkering a vessel or to the operation of unbunkering and feeding the burners.

As regards the economics of moisture, it must be remembered that this must be evaporated either in an external drier or in the furnace. In the latter case, the extra power required in the pulverizer has to be paid for in addition to the inconvenience of choked feeders and irregular firing.

As already stated, it is incumbent that the period taken for the combustion of a particle be as short as possible. After leaving the burner mouth any moisture must first be evaporated at 100 degrees C. The volatiles will not commence to distill until the particle is raised in temperature to 210 degrees C. Heat must be absorbed from the furnace wall or from the flame to evaporate the moisture. Thus a time lapse must ensue from the entrance of the particle into the furnace to the time evaporation is completed. As the atom is probably traveling at the rate of 65 feet per second, it will be understood that in marine work the place to extract the moisture is external to the furnace.

Preheated Air.—If the primary or air accompanying the coal and used for projecting it into the furnace is cold, some time is required for its temperature to increase, since it is sluggish in absorbing radiant heat. The fuel dust absorbs radiant heat at a rapid rate, and parts with some of this by convection to the primary air which lags behind the particles in temperature.

In order, therefore, that particle, gas envelope, and air may all be near the ignition point on entering the furnace, it is desirable to preheat both primary and secondary air. No loss will ensue, since all volatiles and gases liberated from the fuel will enter the furnace. It is preferred, if possible, to obtain a preheat of 230 degrees C. by using a heater placed in the path of the funnel gases.

Turbulent Combustion.—Just as turbulence has been found to play an important part in the combustion of oil in a Diesel engine, so it is a primary necessity for a short-flame powdered-fuel burner.

Experiments carried out by the Naval Board, Commonwealth of Australia.—In 1915 the author, seized with the fact that up to that time no well oil had been discovered in Australia, and that the country was entirely dependent on external sources for the supply of this vital necessity, commenced experimenting with powdered coal for combustion in the boilers of steamships. No data were available for most of the problems that presented themselves, though various firms in America had done a considerable amount of work on long-flame burners. In consequence, figures for stowage, flow through orifices, sieving and effect on flame of different mesh products, friction of pipes, quantity of air required to convey coal, and velocity of air to avoid deposition had to be determined before serious work could be commenced.

In 1917 a small quantity of powdered coal was successfully burnt in the Durr boilers of H.M.A.S. *Encounter,* and in consequence the Naval Board,

Commonwealth of Australia, authorized the construction and installation of a plant on H.M.A.S. *Sealark*. This vessel was fitted with cylindrical return-tube boilers. She was equipped with a 60-volt dynamo; and as suitable electric motors could not be obtained for the operation of the powdered-fuel plant, it was decided to securely moor the vessel to the wharf, and rotate the propellers. No powdered-fuel plant was available, so powdered fuel of the desired fineness was purchased and slowly dried in coke ovens to from 1 to 2 per cent of moisture, except in the case of Morwell brown coal which contained considerably more.

The problems to be solved were largely based on warship requirements, and were:—

1.—Absolute reliability and immunity from spontaneous combustion or explosion.
2.—Fuel to be taken aboard with facility free from fire risk.
3.—Fuel to be carried in bunkers for an indefinite period.
4.—Fuel to be removed mechanically from bunkers of irregular shape.
5.—Sufficient fuel to be carried in reserve in stokeholds for a limited period of steaming, in case of a defect in the trimming machinery.
6.—The stokeholds to be free from dust, and all pipes, joints and conveyors to be dust-proof.
7.—The fuel to be burnt with at least the average efficiency of hand-firing.
8.—Means to be provided for readily removing the ash.

The furnace first used was of the ordinary Morison corrugated type, lined with brickwork. The back of the combustion chamber was similarly lined and a chequer-work was built across the furnace towards the front end. Natural draught was employed for the secondary air and forced draught for the primary air.

Sufficient allowance was not made for the volume of the gas and friction, so the chequer-work was removed and a wall with an opening of 3.5 square feet substituted. The brickwork surrounding the furnace beyond the wall was also removed and the furnace extended into the stokehold 2 feet.

Subsequently the boiler furnace was altered to the shape shown and an unofficial trial run, which gave quite favorable results.

Test No. 3 was of short duration; South Clifton coal—12,402 B.T.U.'s per pound—was used. It will be noticed that the furnace proper was entirely cleared of brickwork and that the combustion space volume was increased to 360 cubic feet by the addition of external firebrick chambers. This enabled a larger quantity of coal to be dealt with. 850 pounds of coal per hour evaporated 7,660 pounds of water from and at 100 degrees C., the rate being 2.36 pounds per cubic foot of total furnace or 4.2 pounds per cubic foot of the original boiler furnace, the latter being 52,000 B.T.U.'s per cubic foot and the efficiency 72.4 per cent.

The following extracts are taken from the official report of the supervising officers, which was issued after the trial:—

"The recent experiments have, it is considered, proved the following points:—

1.—That powdered coal can be burnt satisfactorily as compared with other fuels for warship purposes.
2.—That it can be conveyed continuously and efficiently from a ship's bunker to the furnace, and the combustion and smoke production regulated as easily as with oil fuel.
3.—That the fuel can be delivered into the bunkers easily and efficiently.
4.—That the conveying machinery, fuel supply, etc., can be installed at comparatively low cost, without undue loss of power or excessive weight."

Storage of Pulverized Fuel.—The solid particles of powdered coal have a specific gravity of from 1.32 to 1.35, and the volume occupied by one ton of coal after being passed through free air is about 54 cubic feet per ton. The percentage of aeration falls rapidly to 50 cubic feet per ton, and stored in bunkers subject to a certain amount of vibration the volume drops to 46 cubic feet per ton, which may be accepted as an average figure, and was ascertained by the author after storage for seven days in a ship's bunker.

In order to secure freedom from risk of spontaneous combustion or explosion, the author, in the system of pulverized fuel firing which he has evolved, observes the following precautions:—

The fuel is dried and ground on shore. The grinding station may be on the foreshore, or half a mile away. No difficulty is experienced in forcing the material through pipes. The propelling agent is inert flue gas from the drier, which has been cooled and compressed to a pressure of 40 pounds per square inch. This gas contains CO_2, nitrogen, and a certain proportion of oxygen and moisture, the latter greatly reduced by compression and subsequent cooling.

The quantity of gas required is 5 cubic feet at 40 pounds pressure per ton of fuel.

The ship end of the hose is connected to a gas-relief separator fastened to the bunker lid. Before bunkering, a quantity of gas is blown into the bunker through the hose, this displacing any air present.

One ton of dust per minute may be forced through 500 feet of 4-inch piping by a pressure of 40 pounds per square inch.

Each particle of coal is sealed by inert gas, with which all the interstices are filled, to the exclusion of oxygen; consequently no vehicle exists for the propagation of flame.

CO_2, being heavier than air, displaces the latter, which flows away through the separator.

On completion of bunkering the bunker-lid joint is made, and the bunker sealed to prevent admission of air and its diffusion into the CO_2.

Since the compartment is airtight, no more moisture can gather than was in the compartment when empty, except for such as is contained in the fuel dust and gas content.

Emptying Bunkers.—This proved one of the most difficult problems to surmount, as the system had to be flexible enough to remove fuel from several spaces of irregular shape, and at the same time be dust-tight. Screw conveyors and suction pipes into the dust were tried. The former were not flexible enough for large compartments of irregular form, and the dust bridged or flooded, giving uncertain feeding. The suction pipes blocked and required too many inlets. Eventually a solution was obtained by leading pipes through the bunker, the inlet end being led to warm air near the boiler uptakes and fitted with a cock worked from the stokehold plates. The other end of the pipe was secured to a fan capable of maintaining a velocity flow of 80 feet per second. The weight of air required to convey 1 pound of coal-dust without deposition is 3½ pounds. Ports were cut in the pipe within the bunker, and these were fitted with a cover-plate worked by a lever from the stokehold. The fan discharged to a dust separator, which is fitted in its upper part with a special type of large area filtering medium, capable of shaking itself free of adherent dust. A ready-use bin, capable of holding sufficient reserve fuel for four hours' steaming, was situated below the separator and received its effluent.

Partial success was achieved; and further consideration being given to the problem, it was determined to restore the fluidity of the dust by injecting

inert funnel gas, thus breaking the partial vacuum formed by the evacuation of the coal and restoring the aeration to normal. Success followed this innovation, and the dust flowed regularly without sticking or bridging.

As before mentioned, funnel gas is abstracted, cooled, compressed, and stored in a reservoir. The pipe from the latter is led into the dust at a few points, and by opening the cock at an interval of two hours for a few minutes trouble is averted. This is called the fluffing system, and by its use powdered fuel will flow from a 'tween deck to a lower bunker. The pressure used is slightly above that due to the head of coal.

Ready-use Bin.—This is fitted to provide storage for sufficient fuel for a limited period of steaming, and also as a settling space for the material received from the separator above. The bin is dust-proof.

Feeder.—Consists of a casting secured to the bottom of the ready-use bin, the access to which is controlled by a dust-tight valve. It has also an inlet for primary air, and an exit for dust and air mixed.

A machine-cut screw feeder, having one end housed for a sufficient length to prevent flooding, rotates in a casing and feeds the dust forward to the down pipe. The screw is manipulated by a variable-speed gear, motor driven. The feed of coal requires a range of from 60 pounds per hour up to a maximum for a large furnace of 700 pounds per hour, and is designed in ranges of 300, 500 and 700 pounds. The horsepower required for these is 0.5, 0.75 and 1 per feeder.

Carburetor.—The function of this is to intimately mix the dust and air and prevent the fuel dropping down the center of the pipe. The carburetor is fed with hot primary air. It consists of a removable diaphragm of 80-mesh gauze wire of sufficient total area to pass all the air and coal with allowance for frictional loss. The diaphragm can be rotated half a turn and bumped against a stop; this is to enable material that will not pass the mesh to be shaken off into the furnace.

Burner.—This consists of a W.I. tube, whose exit end is conically expanded; into the extra area thus formed a C.I. cone of a different angle to the pipe is inserted. The cone is secured to one and a half turns of worm, of pitch about one and a half times the diameter of the pipe. The worm is fastened in the pipe. The other end of the burner pipe is fitted with a handle, and terminates in a gland packed with one turn of asbestos. The burner pipe slides on the pipe leading from the carburetor.

Air Cone.—The function of this is to provide the secondary air required, and to impart to it a tangential motion. The cone is built of cast iron, and one end of it is recessed to carry a ring of firebrick 1 to 1¼ inches thick. Just behind these in an axial direction the air vanes commence; these are given an angle of 45 degrees. The back of the cone is kept away from the air casing, and a certain volume of air flows through this space; this provision ensures that all the dust is projected into the furnace and that none falls into the air box.

Air Box.—This is secured to the furnace extension, and receives the secondary air pipe, the latter being fitted with a control air valve.

Extension Furnace.—This is built out from the furnace, and in the design provision is made for air circulation outside the brickwork, air-cooled bridge, door at bottom of furnace proper for clearing dust, ash door to the furnace extension to permit access to the back end or brickwork, and to provide for clearing ashes or slag. The extension also allows of the smoke-box doors being opened, and of a walk way below it for the attendant. Its shape also gives the necessary changes of direction and velocity to deposit slag particles.

Operation of the Boiler.—The ready-use bin having sufficient coal in it, the fan is started at low speed, and air admitted to the primary pipe. A lighted torch made of oily waste is passed through the spectacle hole to the front of the burner. The latter is drawn back to the inner edge of the cone brickwork.

The feeder screw is then started at lowest speed; no secondary air is required. At the end of five minutes the torch can be withdrawn. In 15 minutes the cone bricks will be sufficiently warm to support combustion, and the feeder pipe can be advanced further toward the furnace. It is inadvisable to exceed a feed-rate of 60 pounds per hour until the refractory in the furnace extension reaches a temperature of about 500 degrees C.

The primary air supply will be sufficient till an input of 100 pounds of dust an hour is reached; then secondary air should be admitted. As soon as the brickwork is incandescent and the temperature of the wall of the furnace proper reaches about 120 degrees C., the primary air can be opened out and the secondary air regulated for the input of dust and power required.

Further Tests with Marine Boilers.—Messrs. Clarke, Chapman & Co., Ltd., have interested themselves in the question of pulverizers, pulverized fuel, and the use of this fuel in Scotch boilers at sea. They have carried out a number of trials with different types of burner and carrying arrangement of furnace front.

In the first attempt, in which they unknowingly followed closely on the lines of discarded experimentation in both Australia and the United States, the usual bricked-up furnace and burner at the furnace mouth appeared. Up to 350 pounds of dust per furnace was consumed, but a considerable amount of partially unburnt fuel was carried through the tubes into the smoke-box. Ignition of this unburnt fuel and carbon monoxide took place in the smoke-box. For the next stage, the furnace brickwork was reduced, the furnace mouth extended, and the hanging bridge left in position. Tests showed that 400 pounds of fuel could be burned per hour with a carry-over to the smoke-box, the temperature of which was 400 degrees C. A furnace temperature of 1,400 degrees C. was recorded. For the next step all the furnace brickwork was removed and the furnace further extended. Opposed burners were introduced, also induced secondary air, which is heated in a casing around the brickwork.—"The Shipbuilder," August, 1927.

ALUMINUM BRONZE.

Bronze aluminum is an alloy entirely French, discovered by a French scientist. More tonnage is produced in France than any part of Europe.

Seventy years ago Mr. H. Sainte Claire Deville ascertained remarkable properties of this alloy composed of 90 per cent copper and 10 per cent aluminum, having a mechanical resistance equal to steel. It offers the advantage of being practically unoxidizing, even at high temperatures. It resists corrosion of numerous chemical agencies. In particular it resists salt water better than any other alloy. It has been verified by the English who have made castings of aluminum bronze and subjected them to a year's immersion in different waters, resulting in a perfect resistance to corrosion.

Aluminum bronze has remained practically unused for many years. In 1909, Mr. H. Le Chatelier was astonished to see its limited use, recalling the advantage of its application in marine construction, principally propellers and all bronze pieces demanding a strong chemical and mechanical resistance. Where the use of steel is a serious question he has constantly

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urged the use of aluminum bronze even in existing work in all machines where rust may develop, if only for the conservation of material. Until recently the prevention of shrinkage in pouring the aluminum bronze has proved very difficult; there is considerable shrinkage on solidification, release of gases, and above all, difficulty in elimination of alumina.

In the preparation of an alloy, oxides must be avoided to insure a good article. Aluminum is an energetic reducer; its addition to other metals has the effect of transforming into alumina all the oxides contained in the bath. If this alumina remains in the alloy it radically spoils the product. If we succeed in excluding the alumina we at the same time carry off all the oxides. In that alumina is nothing but the residuum, we obtain an alloy whose purity is perfect.

In the Durville process the result is accomplished as follows:

The molten metal is placed in a runner "P" attached to ingot mould "L", position (1). The alloy is at a high temperature and the oxides transformed into alumina come to the surface. The dross is skimmed but the coating of the alumina instantly recurs when in contact with the air. The metal in the runner is transferred to the ingot mould "L" without agitation and in such a manner that the coating of the alumina constantly protects the molten metal against further oxidation due to the bath's invertion.

In this manner the ingot mould "L", the runner "P" and the communicating duct "C" are arranged so that these three elements are in a straight line during the revolving of the system of 180 degrees. In this manner the alloy passes from the pocket in the ingot mould position (2). The level of the metal bath is always in a horizontal plane, the pocket making this effectual. There is no agitation; the shallow skin of the alumina in which the surface varies in the course of the operation insures a complete protection of the metallic mass against contact with the air. At the end of the operation (3) the alumina is entirely in the upper part of the lump that is detached. It goes without saying that the details thoroughly studied will permit the reduction to a minimum of the top scrap, due to great shrinkage of the metal localizing a small volume for top discard. Aluminum bronze ingots prepared by the Durville process are sound and homogeneous, and can be forged, rolled, etc., into bars, plates, tubes, wire, forged pieces, castings, etc.

This metal offers a wonderful resistance to repeated shock. A Durville bronze bar of 130,000 pounds per square inch strength withstood 3,181,000 shocks, while a bar of steel of the same size withstood only 522,520.

The Merchant Marine is already using aluminum bronze for replacing parts liable to corrosion. It is used in torpedo details, eccentrics, friction plates, turbines, gyroscopes, many uses in submarines and Diesel motors, etc. The possibility of obtaining an economical aluminum bronze of all forms such as used in aviation, automobiles and marine work, must be through its general use. Trouble so often experienced in condenser tubes is abated in the use of aluminum bronze. This alloy is equally good when either exposed to great heat or salt water. All marine boilers using copper should be replaced with aluminum bronze, because the tensile of plates of this metal is from 90,000 to 100,000 pounds per square inch. This material can be welded or brazed without loss of tensile strength.

Aluminum bronze plates are used for aviation hulls and reservoirs, which permits reduced thicknesses of metal, less weight than duralumin, and a greater resistance.

Bronze of 10 per cent aluminum possesses the color of gold. Very early this attracted the attention of Mr. H. Sainte Claire Deville. He started making utensils which have been in service sixty years and still maintain their brilliancy as if made today.—"La Revue Maratime."

DIESEL-ELECTRICITY FOR NEW CONVERSIONS.

Diesel engine builders throughout the United States will welcome the Shipping Board's recent, though it must be confessed rather tardy, step for the Diesel-electrification of two of three existing steam freighters.

The success with which the existing Shipping Board Conversions are operating, as we point out in the first article in this issue, is a plain indication of the fact that other ships can be converted and can operate with equal success. If the American mercantile marine is to remain on the high seas, and if, as we have shown, the Shipping Board is keeping it there, then obviously we must have more conversions * * * and quickly.

The ships which it is proposed to convert are exceptionally fine shelter deckers of nearly 12,000 tons deadweight, which should lend themselves to conversion with a minimum of cost. At present they have turbines of 3000 shaft horsepower which give them a sea speed of about 10.5 knots. The ships have an overall length of 457.5 feet, a beam molded of 56 feet and a depth molded of 38 feet. The forward and aft ends of the ship will be removed and slightly lengthened and a new type stern substituted which is expected to give the ships an increased speed.

The Shipping Board has authorized Commissioner William S. Benson to employ Gibbs Brothers, Inc., of New York, and Rear Admiral D. W. Taylor, U. S. N., Ret., jointly, to prepare plans and specifications under which bids will be sought for alteration and installation of Diesel electric equipment on one or two of three large steel cargo steamers now laid up.

Admiral Taylor and William Francis Gibbs, of Gibbs Brothers, have been working together on experiments relative to the possibilities of increasing the speed of ships by alteration of the hull line. They have submitted a report in which it is stated that their investigations indicate that certain changes in the design of the vessels considered will increase their speed.

The ships under consideration are the *Courageous, Defiance* and *Triumph*, each of which is 11,773 tons deadweight.

Up to the present time the ships converted to Diesel propulsion have been for operation on direct Diesel drive. The plan for these two ships will call for Diesel electric drive with the main motors in the present engine room.—"Motorship," August, 1927.

GERMAN DOUBLE ACTING DIESEL ENGINES.

New Types of Diesel Engines Which Are Displacing Single Acting Machinery in the German Shipyards.

Generally speaking, it is perhaps true to say that more interest is being concentrated throughout the whole world on the development of the double acting engine than of the single acting type which most engineers appear to consider to have reached its limit; but it is in Germany alone that double acting motors are already being built in larger numbers than those of other types.

The course of development of marine oil engine design in Germany is worthy of examination. Two-cycle and four-cycle, single acting machinery was built before the war, and the first double acting, two-stroke engine ever fitted in a ship was constructed and installed in Germany just before the outbreak of war. After the conclusion of hostilities, however, all the leading manufacturers appeared to think that as most experience had been gained

with the four-stroke, single acting type, it was better to build such engines and the large firms such as Krupps, the M. A. N. and the A. E. G. worked along these lines. In fact, at the A. E. G. in Berlin, four-cycle, double acting Burmeister and Wain engines were standardized on a very extensive scale.

About two years ago, the double acting, two-stroke engine came into prominence, and so rapidly has it grown in favor that there are more ships, both passenger and cargo, now under construction in Germany fitted with such plant than vessels equipped with any other class of internal combustion machinery. The Hamburg Amerika Line in particular has shown its preference for double acting motors. With largely increased capital it has inaugurated a building program involving the construction of about 22 new ships of which 18 are to be propelled by internal combustion motors. Of these, 12 or 14 are to have double acting units, three being of the new A. E. G.-Hesselman type and the remainder the M. A. N. design.

The first ship under this new program has just been completed. It is the cargo carrier *Rheinland* intended for the Australian service, and carries 10,100 tons deadweight, being 450 feet in length, with a beam of 59 feet 11 inches. The machinery installation is of exceptional interest, for it indicates the attention which some owners are paying to the possibilities of standardization in Diesel motor construction.

It is a 6-cylinder M. A. N. double acting, two-stroke type, designed to develop 4500 brake horsepower at 86 revolutions per minute and having six cylinders, 600 millimeters bore with a piston stroke of 1400 millimeters. Nine more of the new Hamburg Amerika Line cargo or mixed cargo and passenger ships are to have similar machinery, but owing to the variation in size and speed, the power needed ranges from just under 4000 up to 5400 shaft horsepower. In order to retain the same cylinder size throughout, however, it was decided to adopt similar engines but with different numbers of cylinders, with the result that, in some cases, 5-cylinder motors have been used, in others 6, and in two vessels, 7-cylinder plant will be installed. In their various ships, therefore, the owners will have engines totalling over 50 cylinders, identical in all respects. The advantage which thus accrues in connection with the question of spare parts and overhaul is obvious, while engineers can readily be taken from one ship to another, and remain completely familiar with the machinery.

For three of their new vessels, the Hamburg Amerika Line have ordered double acting, two-stroke motors of the A.E.G.-Hesselman design which has not yet been fitted for marine work. This would seem to indicate that little hesitation is now shown in adopting Diesel motors of new types, although it should be added that prolonged experimental work was carried out in the A. E. G. factory at Berlin when the order was placed.

The new motor will be the first double acting, two-stroke unit to have airless injection (the Richardson-Westgarth type has not yet been installed in a ship) and remarkably low fuel consumptions are claimed. In each of the three ships mentioned, there will be a 6-cylinder motor developing 4500 brake horsepower at 86 revolutions per minute and, incidentally, it may be noted that practically all the new cargo tonnage for the Hamburg Amerika Line is represented by single-screw vessels with a machinery speed of no more than 85 to 90 revolutions per minute—a decidedly low figure for Diesel motors.

The machinery of the two 17,000-ton passenger liners *St. Louis* and *Milwaukee* which are being built for the Hamburg Amerika Line's New York service, with a speed of 16½ knots, is of peculiar interest, and some particulars of this have already been made public. There will be four double

acting, two-stroke motors in each ship developing a total of 12,400 brake horsepower and running at 230 revolutions per minute, but they will gear on to two propeller shafts turning at 110 revolutions per minute. It is a curious fact that while there will soon be five large German passenger liners in service equipped with geared Diesel machinery, no shipowner in any other country in Europe has thought fit to adopt this arrangement, which, if successful, has many advantages.

In addition to the A. E. G.-Hesselman and the M. A. N. designs two other types of double acting, two-stroke engines are now being built in Germany. No attention is, however, being devoted to double acting, four-stroke motors which are the most popular in the United Kingdom and Scandinavia. The interest of the German Government in this problem may be inferred from the production of a double acting, two-stroke engine at its old Naval Dockyard at Kiel, which now forms the Deutsche Werke, and in which the German Government has a big financial, if not a controlling, interest.

An illustration of the Deutsche Werke experimental motor is given, and trials are about to be run of the first 6-cylinder engine of this class. Four are under construction, for two twin-screw cargo ships for the Norwegian owner, Ivar An Christensen, both of which, it is claimed, will attain a speed at sea of 16 knots. The engines are designed to develop about 4000 brake horsepower each in six cylinders, 650 millimeters bore with a piston stroke of 900 millimeters and they run at the relatively high speed of 125 revolutions per minute. They use air injection and scavenging air is supplied from separate electrically operated blowers.

The remaining double acting engine of German design is that of Krupps, which has, however, only been developed in the experimental stage. The single cylinder unit is capable of an output of about 1800 indicated horsepower, or approximately 1400 brake horsepower. It includes no unusual features in design, except that, as a result of recent investigations, it has been decided that a hydraulic system of operating the fuel valves presents advantages over the normal arrangement in which the cam shaft is utilized for this purpose.

Taking a survey of the present position of marine Diesel engine manufacture in Germany, the conclusion must be reached that the double acting, two-stroke engine bids fair to gain almost universal application for moderate and large powers, while for smaller units, say up to 1000 horsepower, the new airless injection four-cycle, single acting type, now built by several firms, will be widely used. It must be admitted, however, that a number of two-cycle, single acting engines of the Sulzer, Krupp and M. A. N. types are still being built and it may be that, after gaining further experience, owners will yet revert to them on a larger scale.—"Marine Engineering and Shipping Age," October, 1927.

EXTRACTS FROM THE GERMAN PRESS.

EXPERIMENTS WITH NEW SHIP LINES BY MAIER.

Exhaustive experiments have been conducted at the Hamburg Marine Experiment Station on the effect of the so-called Maier ship lines. These ship lines were proposed about 20 years ago by an Austrian engineer, Maier, and are characterized by a peculiar shape of forward and aft frames. These are so designed that the centers of gravity of half the frame areas lie on a

very flat curve and the lower part of the frame is thereby cut to a triangular form. This results in shortening of the path of the water flowing along the ship's body and as a consequence reduces friction and eddy losses.

These experiments, conducted on behalf of Deutsche Shiffs-und Maschinenbau-A. G., showed for a fishing vessel of 570 tons displacement and 10 knots speed a 19 per cent reduction in horsepower required to propel it. A small passenger vessel of 300 tons displacement and 22 knots speed showed a 25 per cent reduction in horsepower. A freighter of 4500 tons displacement and 10 knots speed gave a 16.67 per cent, and a larger freighter of 18,300 tons and 14 knots speed showed 15.5 per cent reduction in horsepower, whereas a large passenger liner of 16,500 tons and 27 knots could be propelled by 11 per cent less horsepower. All ships used for comparative purposes had good normal lines.

Models with the new lines were also tested for seaworthiness, and they proved much quieter on the waves as compared with those of the normal ships and at high speed they would take no water over the forward part of the ship.

The firm sponsoring the experiments has several ships with these new body lines now under construction and expects to equip them with steam installations of Bauer-Wach type, using exhaust from the reciprocating engine in a high-speed turbine geared to the engine shaft. The results so far obtained with this type of propelling plant show that with the same amount of fuel used 25 per cent more shaft horsepower may be developed, or for the same shaft horsepower a 20 per cent reduction in fuel consumed obtained.—"VDI", 1 October, 1927.

TWO-CYCLE VS. FOUR-CYCLE, OR *AORANGI* VS. *GRIPSHOLM*.

By S. Bock.*

Operating records of the two large mail and passenger motor liners—*Aorangi*, operated by the Union Steamship Company of New Zealand, and *Gripsholm*, owned by Swedish-America Line—have recently become available and invite comparison between the capacities of the engines and their fuel consumption in actual operation. These comparisons are of especial interest because both ships are very much alike in size and speed, but have different propelling plants—namely, two-cycle single acting engines are installed in the *Aorangi*, and four-cycle double acting machinery is used in the *Gripsholm*.

The principal dimensions of the ships as well as engine data are compiled in the following table:

Ship	*Aorangi*	*Gripsholm*
Length, feet	580.1	550.0
Beam, feet	72.2	74.0
Depth, feet	46.6	42.6
Tonnage—Gross	17,491	17,300
Deadweight	23,300	23,600
No. of propellers	4	2
Type of engine	4 single acting two-cycle, six-cylinder, Sulzer	2 double acting four-cycle, six-cylinder, B. & W.
Total shaft horsepower	13,000	13,500

* Translated by E. C. Magdeburger from "Schiffbau," 18 May, 1927, p. 231.

Fairfield Shipbuilding and Engine Co., of Glasgow, built the Sulzer engines of the *Aorangi*. These are of 27.5-inch bore, 39.0-inch stroke, and normally operate at 120-125 R.P.M.

The four-cycle double acting engines of the *Gripsholm* were built by Burmeister & Wain Company of Copenhagen and have 33.07-inch cylinder bore, 59.05-inch stroke, and run at 115-120 R. P. M. The *Gripsholm's* engines have separately driving injection air compressors in order to increase their mechanical efficiency and in this manner reduce the cylinder bore, whereas the *Aorangi's* engines have direct-driven air compressors.

On test block the mechanical efficiency of the Sulzer engines proved to be .822, whereas the B. & W. double acting engine had a mechanical efficiency of .7924 under their respective full-load conditions—3250 B.H.P. and 6750 B.H.P.

The *Aorangi* has been in continuous service between Vancouver and Australia for the last two years. Its propelling plant has proven very satisfactory and during the first year of operation a distance of 100,000 miles was covered. The mean speed at sea is between 17.0 and 17.25 knots, the mean shaft horsepower developed is 11,400 S.H.P. at 118-120 R.P.M. The mean fuel consumption per day for all purposes proved to be 58.5 tons, or .472 pound per S.H.P. per hour of the main engines.

A western trip(Göteborg-New York) of the *Gripsholm* under average weather conditions gave the following average values: Speed, 16.05 knots at 113 R.P.M.; shaft horsepower—12,400 S.H.P. (15,740 I.H.P.), with the best speed outside of the Gulf Stream—16.45 knots. A good eastern trip (partly assisted by the Gulf Stream, gave a speed of 16.75 knots when developing 12,530 S.H.P. (15,860 I.H.P.), with the best speed of 16.95 knots, showing that speed was very constant. Thus the design average speed at sea of 17.0-17.25 knots was not maintained and rumors spread that the *Gripsholm* can not develop her normal full power of 13,500 S.H.P. due to severe vibration that became manifest at full speed. A difference of .5-.75 knots is quite important for a mail and passenger liner on long voyages.

Fuel consumption of the *Gripsholm* in tons per day is reported as follows:

	Westbound.	*Eastbound.*
Main engines	47.2	48.3
Auxiliaries	10.4	9.4
Galley, heating, etc.	7.15	8.3
Total	64.75	66.0

Thus the main engines alone used—

Westbound—.3485 pound per S.H.P. per hour, or .276 pound per I.H.P. per hour.

Eastbound—.355 pound per S.H.P. per hour, or .280 pound per I.H.P. per hour.

On test stand when carrying full load the fuel consumption was .3815 pound per S.H.P. per hour and .302 pound per I.H.P. per hour using exactly the same kind of fuel as in actual service. This considerable difference between .355 and .3815 pounds needs in itself an explanation. The horsepower required to drive the auxiliary machinery has an average value of 1750 B.H.P. and the above figure of daily fuel consumption gives .544 pound per B.H.P. per hour. The operating auxiliary engines usually run as near full load as possible to take advantage of improved fuel consumption. The rating of the auxiliary engines of the *Gripsholm* is normally

170-190 B.H.P. per cylinder and their fuel consumption should by no means exceed .44 pound per B.H.P. per hour. It would appear, therefore, that of the fuel consumed, too much is credited to auxiliary engines and not enough to main engines.

Fuel consumption for all purposes referred to the shaft horsepower developed by the four-cycle main engines of the *Gripsholm* figures out at .481 pound per S.H.P. per hour westbound, and .483 pound per S.H.P. per hour when eastbound.

The two-cycle machinery of the *Aorangi* develops about 1000 S.H.P. less and gives the ship on the average .5 knot more speed than the *Gripsholm*, using .472 pound of fuel per S.H.P. per hour for all purposes. Thus the two-cycle engined *Aorangi* has a better fuel consumption than the four-cycle engined *Gripsholm*. The displacement of the two ships is very much alike, as shown above, so that the two-cycle ship with its four propellers shows a better propulsion efficiency and uses besides 7 tons less fuel per day than the four-cycle ship.

The main engines of the *Aorangi*, developing an average of 11,400 S.H.P., use 48 tons of fuel daily, or .386 pound per S.H.P. per hour. This is very good indeed considering that two air-compressor cylinders are driven from the engine shaft. The scavenging air is furnished by motor-driven centrifugal blowers, requiring about 700 B.H.P. to be developed by auxiliary engines using about 309 pounds of fuel per hour. Since scavenging air of necessity belongs to the two-cycle engines and the main engines themselves use 4410 pounds of fuel per hour, a total of 4719 pounds of fuel per hour must be credited to the main engines, which referred to 11,400 S.H.P. developed, gives a fuel consumption of .415 pound per S.H.P. per hour.

For the *Gripsholm* a fuel consumption of .3815 pound per B.H.P. per hour as determined on test stand must be assumed as more correct, which gives for an average of 12,500 S.H.P. developed a fuel consumption of 4760 pounds per hour. To this must be added the fuel consumed in developing 1200 B.H.P. in auxiliary engines furnishing injection air to the main engines of the *Gripsholm* in order to arrive at the same basis of comparison. Assuming .44 pound per B.H.P. per hour a total of 528 pounds per hour is consumed by auxiliary engines, giving a total of 5288 pounds to be credited to the main engines or .424 pound per S.H.P. per hour.

Thus the corrected figures compare as follows:

Single acting two-cycle—.415 pound per S.H.P. per hour.

Double acting four-cycle—.424 pound per S.H.P. per hour,

showing that the fuel consumption of the two-cycle engine is fully equal to that of the four-cycle type.

Comparison of these figures with the results available on Sulzer engines both of two-cycle and four-cycle type installed in ships of medium speed (10-14 knots) gives a similar result. In fuel consumption per S.H.P. per hour both types of engines are usually equivalent, contrary to the statements made by advocates of the four-cycle type. Other independent data now available would confirm that also. However, a slow speed freighter equipped with two-cycle machinery has a decided advantage in speed over the one equipped with four-cycle machinery. The lower speed of the two-cycle engine—85-90 R.P.M. as compared with 110-130 R.P.M. of the four-cycle engine gives better propulsion efficiency and consequent better fuel consumption per mile—and that is really the only thing that concerns the shipowner.

GIANT WHITE STAR MOTOR LINER.

"Schiffbau" of 7 September, 1927, p. 391, reports the beginning of construction on a Diesel engine propelling plant for a giant motor passenger liner ordered by the White Star Line from Harland & Wolff, of Belfast. This plant will be of 100,000 horsepower and the vessel of not less than 50,000 tons displacement. With the completion of this giant and the two ships now on order for North German Lloyd, not less than eight passenger liners of over 42,000 tons, namely—four English (*Majestic, Olympic, Berengaria* and the new White Star motorship), one American (*Leviathan*), two German (*Europa, Bremen*) and one French (*Ile de France*)—will compete in the North Atlantic passenger service.

WAVE POWER.

By Lybrand Smith,* Washington, D. C.

The two main difficulties which lie in the way of using wave energy are its irregularity and its low quality or concentration, by which is meant the relatively low power available from a cubic foot of water in a wave when compared, for example, to that in a cubic foot of water at Niagara.

The obvious and practical way to overcome the first difficulty is first to receive the energy as it comes from the wave; second, to convert it into *potential* energy in an accumulator, and finally to utilize this energy as required. The obvious accumulator of potential energy in this case is an elevated reservoir of water.

It is to overcome the second difficulty—the low quality or concentration of energy—and the change of form of wave energy from *kinetic* to *potential* energy that is the subject of invention.

REQUIREMENTS OF WAVE MACHINES.

A practical machine for the purpose of extracting energy from sea waves must have the following fundamental characteristics:

(*a*) It must be a rugged structure.
(*b*) It must be a simple form with few moving parts. (Both requirements are obvious to one familiar with the destructive force of sea waves).
(*c*) It must be capable of vertical adjustment for height of tide.
(*d*) It must be capable of horizontal adjustment when, on different days, the waves break at different distances from the shore because of the effect of different tides or winds.†
(*e*) It must convert the *kinetic* energy of the wave (the form in which most of the available energy exists) into *potential* energy with a minimum loss.
(*f*) The receiving end of the machine should be in deep water where the waves have more power; while the operating end should be in shoal water (or on land) where the waves will not cause damage.

The machine described in the following paragraphs meets all of these requirements.

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† This is not to intimate that such a machine should be put in the breakers. On the contrary, it should be kept out of them. The breakers are merely used here as a reference point to explain the necessity for horizontal adjustment.

THE MACHINE.

The chief element of the proposed wave motor is a mechanism known as the hydraulic ram.

Referring to the diagrammatic sketch in Figure 1, A is a hydraulic ram showing the waste valve, the non-return valve, and the air chamber. Leading straight out to sea is a *long* and approximately horizontal drive pipe B. The length of the drive pipe not only permits the ram to be mounted back of protection inshore, while the open end of the drive pipe is in the powerful waves of deeper water, but also increases the efficiency, as it is well known that the longer the drive pipe the greater the water hammer or excess dynamic pressure which causes the ram to lift water.

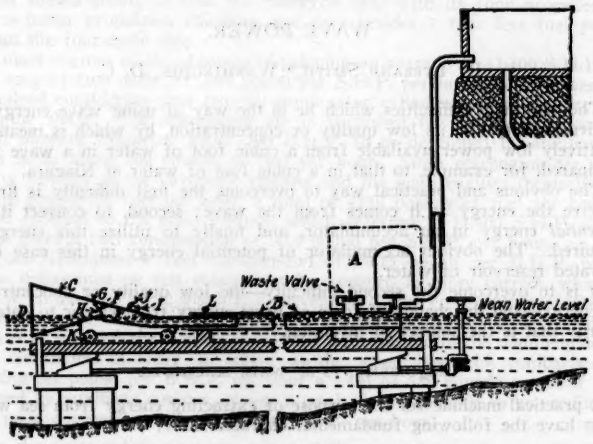


FIG. 1.—DIAGRAM OF A DEVICE FOR USING WAVE POWER.

A flow of water in the drive pipe is set up by the impulse from the waves. Some of this impulse comes from the static pressure due to the height of a wave above mean water level, but this is trivial compared to the dynamic pressure available in the moving water of the waves.

Many people make the mistake of thinking that the water in waves has considerable movement of translation. Actually, except in very shallow water, there is little or no translation of the water. Instead the water particles move in more or less elliptical orbits—the motion in the upper part of the orbit being in the direction of the wave. For example, the velocity of a wave 4 feet high and 96 feet long in water 10 feet deep is 16.7 feet per second, but the velocity of the surface particles of this wave in their elliptical orbits is only 2.7 feet per second, and the velocity of particles below the surface grows less with their depth.

Evidently, if the open end of the drive pipe were exposed to the wave a slight impulse would be given to the water in the pipe, but by mounting a funnel at the end of the drive pipe with its mouth exposed to the waves the

low velocity of about two feet per second will increase. Such a funnel is shown at C in Figure 1 with its open mouth at D. The floor of the funnel is raised at E to form a dam, the purpose of which is to keep the drive pipe completely full of water.

To provide for machines of such dimensions that the inertia of the column of water will not prevent its rebounding seaward over the dam, a rebound, or "flap" valve is provided at F, pivoted at G, and stopped at H. It is believed that this valve could be eliminated where the drive pipe was long enough.

To provide for the escape of imprisoned air a vent is made at I and fitted with a hollow ball J (held in place by a wire mesh) as a valve. Several points are to be noted regarding this air vent and valve:

(*a*) Its direction is such that it will tend to "scoop" the air from the top of the air and water stream.

(*b*) The valve closes by *gravity*, i.e., the ball floats when the funnel is filled by a wave—and this is an advantage, for where the velocity is greatest the static pressure is least. Its action is independent of either static or dynamic pressure.

(*c*) The outer part is such that a wave passing over will not imprison air and force it into the machine. Water might be forced in, but not an appreciable amount of air.

HOW IT WORKS.

Starting with everything at rest, the drive pipe full of water, the waste valve open at the shore end, the air-vent valve open at the sea end, a wave strikes the open funnel mouth. The water particles are prevented from continuing in their elliptical orbits and go forward in the funnel gathering speed as they go. Any air trapped ahead of the wave goes out the air vent until the funnel is full of water; then the valve floats and the vent is closed. Meanwhile, the impulse of this stream or jet through the funnel imparts acceleration to the water in the drive pipe.

Now suppose that the waste valve is adjusted to close suddenly when some velocity v is reached. The question arises, will a very small acceleration produce this velocity? It will if the acceleration is continued long enough—if the time available is long enough. In the case of a wave motor, however, the time intervals will not be of indefinitely long duration, but will be limited to approximately one-half the time intervals between waves. Therefore for a given apparatus it is evident that a wave below some definite force and time interval will not pump water, but neither will it produce a sudden shock or stress on the machine as the energy of such a wave is used overcoming the inertia and friction of the water in the drive pipe.

Proceeding now to larger waves, it is evident, for an apparatus of a given size, that when the waves attain some particular force and time interval, the force will produce the required velocity v in the time interval available, the waste valve will close, and the sudden checking of momentum caused by this closure will produce the water hammer, or excessive dynamic pressure, required to raise some water through the ram to the reservoir.

In the case of a stronger wave with a longer time interval, the greater impulse will cause the water in the drive pipe to attain the velocity v in less time. This greater acceleration, and the longer time interval of a bigger wave will enable the water to attain the velocity v more times per wave. That is, the waste valve will operate more times per wave as the waves grow more powerful and longer.

Every time the waste valve closes, the mass of water in the full drive pipe, its velocity, and therefore its momentum, will be the same. If this momentum is always checked in almost exactly the same time (the time taken to close the waste valve, a fraction of a second), a certain definite water hammer should always result and throw the same amount of water to the same height. The apparatus gains power with more powerful waves by having more water hammers per minute.

How do these facts affect the accumulation of potential energy in this machine? Say the weakest wave which will operate it at all (i.e., produce one impulse per wave) will produce an impulse just sufficient to lift water 106 feet and that the storage reservoir is at 100 feet. This means that each impulse would waste 6 foot-pounds of energy per pound of water; but since the individual water hammers are just the same for a strong as for a weak wave, the storage accumulating efficiency should remain the same through a wide range of wave sizes, the limiting size for maximum efficiency not being reached until a huge storm wave appears which is so powerful that one impulse succeeds another before the former dies out, or until all impulses merge into one, the waste valve being driven tight shut and the water simply squirting through the machine. In such a case the efficiency would decrease. For all practical purposes, however, it may be said that the efficiency of the machine remains at a constant maximum throughout its operation, regardless of variations in wave power.

THE ADJUSTABLE FEATURES.

Normally any wave machine must be capable of adjustment for height of tide. Of course, such machines in use on the Great Lakes could do without this feature and even on some sea coasts the rise and fall of the tide is so slight that the additional cost of installing the apparatus in a manner capable of vertical adjustment might not be compensated by the increased efficiency. Between great rise and fall of tide and no tide would be cases where it might be practical to raise and lower only the seaward end of the apparatus, leaving the ram at a constant level. In such cases the great length of the drive pipe should permit the bending of the pipe through the small angle required without straining it beyond the elastic limit.

Because it depends upon local conditions the method of raising and lowering the machine is purposely shown in the drawing as a crude arrangement of sliding wedges as the object is merely to show that vertical adjustment should be made. Hydraulic jacks, operated by hydraulic power from the storage tank and controlled automatically by a float rising and falling with the tides but protected from wave action, might be used in a large machine.

The horizontal adjustment feature is also merely shown in a simple manner as the actual design will depend on size of machine and local conditions. In the sketch the funnel is shown mounted on rollers K and locked for any particular adjustment by a set-screw L. No means (other than hand power) is shown in the sketch for moving the funnel. Horizontal adjustment of the funnel might be effected by horizontal hydraulic jacks controlled by hand from shore.

Horizontal adjustment is necessary because on different days with different winds and tides the waves break at different distances from the shore. As it is considered inadvisable to have the funnel mouth actually in the breakers, the horizontal adjustment should be provided unless the water is so deep that the waves will never break in the vicinity of the funnel mouth.

POWER AVAILABLE.

If E be the number of foot-pounds per foot of wave front and N be the number of waves per second, then EN is the energy in foot-pounds per second per foot of wave front, and EN/550 is the horsepower per foot of wave front.

When not observed directly, N may be taken from the fundamental wave formula

$$N = V/L$$

where V = velocity, and
L = wave length.

Furthermore, V may be computed from the formula

$$V = C\sqrt{5.123L}$$

in which V is in feet per second, and L is in feet, while C, which is a coefficient equal to the square root of the ratio of the axes of the elliptical surface orbits, may be determined from the following table,

d/L =	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45
C =	0.552	0.746	0.888	0.922	0.958	0.977	0.988	0.994	0.997

in which d is the depth of water, or distance from the center of the surface orbit to the bottom, and L the distance from one wave crest to next. (It is only necessary to use C where the depth of water is less than one-half the wave length.)

The formula for horsepower per foot of wave front thus becomes

$$\frac{E\,C\sqrt{5.123\,L}}{550\,L}$$

As L is a directly observable quantity, it only remains to determine E from the formula for salt-water waves in shallow water:

$$E = 8\,LH^2\,(1 - 19.74a^2/L^2)$$

where E = foot-pounds in one foot of salt-water wave front
L = length of wave in feet
H = height of wave in feet
a = semi-major axis of surface orbits and may be determined in terms of the wave height H from the following table:

d/L =	0.10	0.15	0.20	0.25	0.30	0.35	0.40
a =	0.91H	0.68H	0.59H	0.55H	0.52H	0.51H	0.504H

Substituting this value of E in the horsepower formula, we get

$$\frac{8\,L\,H^2\,(1 - 19.74a^2/L^2)\,C\sqrt{5.123\,L}}{550\,L} = 0.0357675\,CH^2\,(1 - 19.74a^2/L^2)\sqrt{L}$$

as the horsepower* of one foot of wave front of height H, length L, in water d feet deep, the depth affecting the factors a and C.

The length of a wave for any given height, or vice versa, depends upon wind, sea, and complex local influences, but given heights of 2, 4, 6, 8, 10, and 12 feet, the lengths 66, 96, 126, 156, 186, and 216 feet will, under general conditions, correspond to the respective heights sufficiently closely for a practical calculation to show the power available. Using these heights and the corresponding lengths, the horsepower per foot of wave front has been calculated for water 10, 20, and 30 feet deep with the results as shown in Table 1 and in Figure 2.

* The fundamental formulas and tables necessary for the above deductions can be obtained from Merriman's American Civil Engineer's Pocket Book.

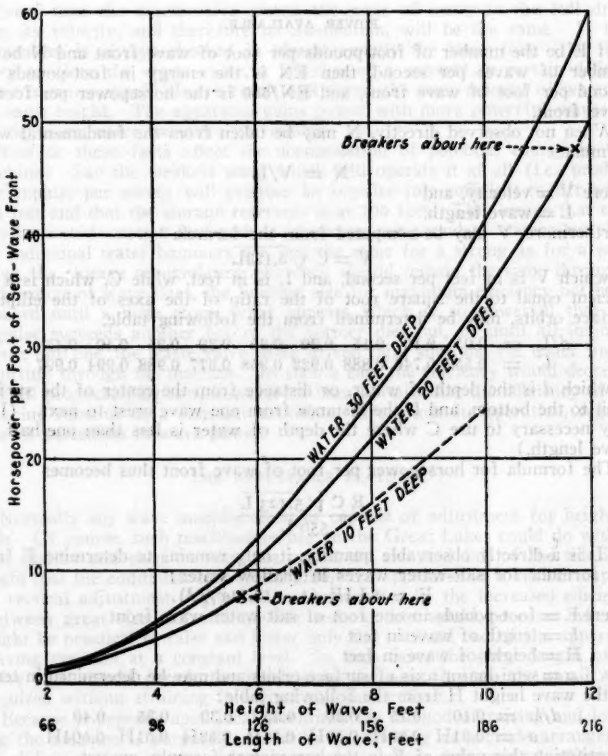


FIG. 2.—HORSEPOWER OF SEA WAVES IN VARIOUS DEPTHS OF WATER.

TABLE 1.—HORSEPOWER PER FOOT OF WAVE FRONT.

Height, ft.	Length, ft.	Horsepower per foot of wave front in water		
		10 ft. deep	20 ft. deep	30 ft. deep
2	66	0.99	1.15	1.16
4	96	4.12	5.15	5.45
6	126	9.29*	12.33	13.58
8	156	16.59*	22.10	25.58
10	186	24.64*	34.96	41.48
12	216	.....	50.66*	61.29

* Breakers.

NOTE: Above values are for waves in *salt* water. For fresh-water waves reduce horsepower by 2½ per cent.

A four-foot wave in water 30 feet deep, a very moderate and usual condition, has 5.45 horsepower per foot of front or 28,816 horsepower per mile, and furthermore, a small increase of height produces a large increase of power.

EFFICIENCY.

While it is more or less idle to discuss the efficiency of a machine prior to its final design, a preliminary idea may be obtained as to whether its fundamental principles would permit a high efficiency in good engineering practice. The probable efficiency could well be discussed under four headings:

(*a*) The mouthpiece (a cone).
(*b*) The drive pipe (a long tube).
(*c*) The hydraulic ram.
(*d*) The accumulation of potential energy.

It is believed that a high efficiency could be obtained in the mouthpiece and that the efficiency should increase with the size of the apparatus when both entrance and exit become larger. Furthermore, where a very long drive pipe is used it is believed that the rebound valve could be eliminated, which would also eliminate loss of head by shock on that valve. Sufficient experimental data are not available to predict accurately the efficiency of the mouthpiece, but considering that most of the energy of a wave is in its upper part and studying data at hand regarding flow through cones, fire nozzles, and jet propellers, it is believed that by good design an efficiency of 90 per cent could be obtained.

The efficiency of the drive pipe would depend largely upon the loss of head in that pipe. As th nature of the machine easily permits a straight drive pipe, loss of head at curves would be avoided. Since the loss of head due to friction is directly proportional to the length of the pipe the loss of head would increase with the length; on the other hand, the dynamic pressure in the drive pipe when the ram operates would increase with the length, and it is believed that the efficiency due to increased dynamic pressure would more than offset that lost through friction due to length of pipe. As loss due to friction is inversely proportional to the diameter of the pipe, the bigger the machine the less the loss, the diameter of the pipe increasing faster than its length. The friction loss increases nearly as the square of the velocity, but the adjustment of the waste valve practically controls the velocity in the drive pipe and therefore the principles permit a maximum efficiency of design, taking into consideration the friction loss due to increased velocity on one hand and the increased dynamic pressure due to increased velocity on the other hand. The friction loss increases with the roughness of the interior surface; good workmanship and good care should keep this loss very low. It seems reasonable to predict that an efficiency of 90 per cent could be obtained in the drive pipe.

From data on the efficiencies of hydraulic rams, it seems not unreasonable to expect an efficiency of 80 per cent for this part of the machine. Merriman mentions a ram used experimentally which had an efficiency of 92 per cent, but does not say by what formula this efficiency was computed.

While the data in the paragraph regarding accumulation of energy were arbitrarily assumed, they are entirely reasonable, and show that the efficiency of accumulating potential energy might well be 94 per cent.

Using these four principal factors in the efficiency of the machine, it seems not unreasonable to predict that approximately 60 per cent of the kinetic energy of waves could be accumulated in the form of potential energy for use

as required in standard hydraulic or hydroelectric machines. Such a prediction, under the circumstances, can hardly be little more than a good guess, but its purpose is to obtain a preliminary idea in regard to the advisability of undertaking further investigation and actual development.

DISADVANTAGES.

Some disadvantages and objections to the proposed machine and the answers to them follow:

(*a*) It is an undeveloped machine.

The purpose of this paper is to interest engineers in the development of wave power. The same objection would apply to any new machine. In view of rising cost of coal, the increasing use of electricity, and the location of the best water-power sites far inland, it is thought that the commercial prospects warrant the development.

(*b*) The first machines would probably have poor efficiency.

The estimated efficiency is more than sufficient to warrant a trial, and even a minor success with such a machine would cause further development by attracting the interest of some of the best engineering brains and skill.

(*c*) The first cost of installation would be high as compared to the first cost of other hydroelectric plants.

A reasonable comparison of the first costs of the two kinds of plants should include every item from the preliminary survey to the cost of the electric transmission lines supplying ultimate consumers of power. These items can well be divided into three groups as follows:

Group 1.

Cost of preliminary survey of site.
Cost of preliminary investigation of power available at site throughout the year.
Cost of site.
Cost of transportation of materials to site and labor of erecting them.
Cost of impounding reservoir.
Cost of electric transmission lines from plant to consuming centers.

Group 2.

Cost of delivery pipes and drive pipes.
Cost of hydraulic rams.
Cost of mouthpieces.
Cost of adjustable features.
Cost of piling on which to mount apparatus.

Group 3.

Cost of standard hydroelectric machines.

Every item of Group 1 is common to both plants. It is believed, however, that an estimate for practically any definite location would show that the cost of every item in this group is *less* for a wave-power plant than for the usual plant. This is particularly true of the costs of transportation and erection, reservoir, and transmission lines, with special emphasis on the last item in the case of supplying power to the large centers of population near the seacoast. The items of Group 2 are an extra charge not imposed on the ordinary power plant. It is probable that they would cost more than the saving which could be effected in Group 1. The item in Group 3 is identical for two equivalent plants and should be considered in making up the total in order to determine the percentage by which the grand total first

cost of one plant would exceed that of the other. It is probable that the cost figures would show a relatively small percentage working against wave-power plants—in their early development at any rate.

(*d*) Exposure to storm waves would damage the machine.

The recent success of using an "air breakwater" to protect exposed harbor works by breaking up storm waves with air bubbles renders this difficulty easy to overcome. The machine itself could furnish the small power required to pump the air under such circumstances.

(*e*) Exposure to ice would damage the machine or interfere with its operation.

Any hydraulic plant exposed to extreme cold would suffer a similar disadvantage. On most of the United States coast there would be little interference from ice.

ADVANTAGES.

The main advantages of this machine are the following:

(*a*) It fulfills the fundamental requirements of a wave motor as stated at the beginning of the paper.

(*b*) Its fundamental principles permit the development of comparatively high efficiency and do not hold the machine to low efficiency (as, for example, in extracting power from coal the fundamental principles of a steam engine hold its efficiency comparatively low).

(*c*) The motive power (waves) is free—which means low operating cost.

(*d*) The power of the installation can be increased indefinitely simply by adding unit after unit, all pumping into the same reservoir. Each new unit can be of latest design without interfering with the others. Multiple units enable one to be overhauled while others are working.

(*e*) The best sites for development are exposed bluffs on the seacoast, which are mainly waste land now. They are consequently cheap.

(*f*) When built near used land such a machine would tend to protect the coast behind it by causing the energy of the waves to do useful work in a machine instead of destructive work on a shore or harbor structure.

(*g*) The first cost would be reduced by easy transportation of materials coastwise as compared to difficult transportation of materials in mountainous regions.

(*h*) It would use water power now wasted, and would use it near localities which need it now.

(*i*) It would not compete with inland water power as the field is broad enough for both without conflict of interests—particularly since the power sites are mutually exclusive.

(*j*) It would not cause competition by tidal power plants, as the best places for tidal power are the worst for wave power, and vice versa. Where both could operate they would be mutually helpful, each one tending to boost the other.

(*k*) Preliminary surveys are unnecessary. If any part of the earth is well charted it is the coast line.

(*l*) Reliable meteorological data are available for all coasts. To determine, for example, the force and direction of the wind prevailing at any place during any month consult a "Pilot Chart" of the U. S. Hydrographic Office. The force and direction of the winds will affect the size and power of the waves. Examine the winds up to 1000 miles offshore. Remember waves approach practically parallel to the coast regardless of direction of wind.

(*m*) The development of such a machine would not conflict with the interests of the large manufacturers of hydraulic or hydroelectric apparatus. Instead it would directly help their interests by affording an increased use for their present standard apparatus.

CONCLUSION.

It is believed that the above discussion suggests the method of building a practical, reliable machine for utilizing wave power. The changing of the suggested method into actual practice is a challenge to competent industrial engineers.—"Mechanical Engineering," September, 1927.

RE-ENGINING THE FRUIT-CARRIER *LA PLAYA*.

In the August issue of last year, we published the decision of the United Fruit Company of Boston, Mass., to re-engine the *La Playa* which, together with her sister fruit-carrying vessel *La Marea*, was built by Camellaird & Co., Birkenhead, England, during the summer of 1923. The *La Playa* and her original machinery installation was extensively described and illustrated in our November and December issues of 1923.

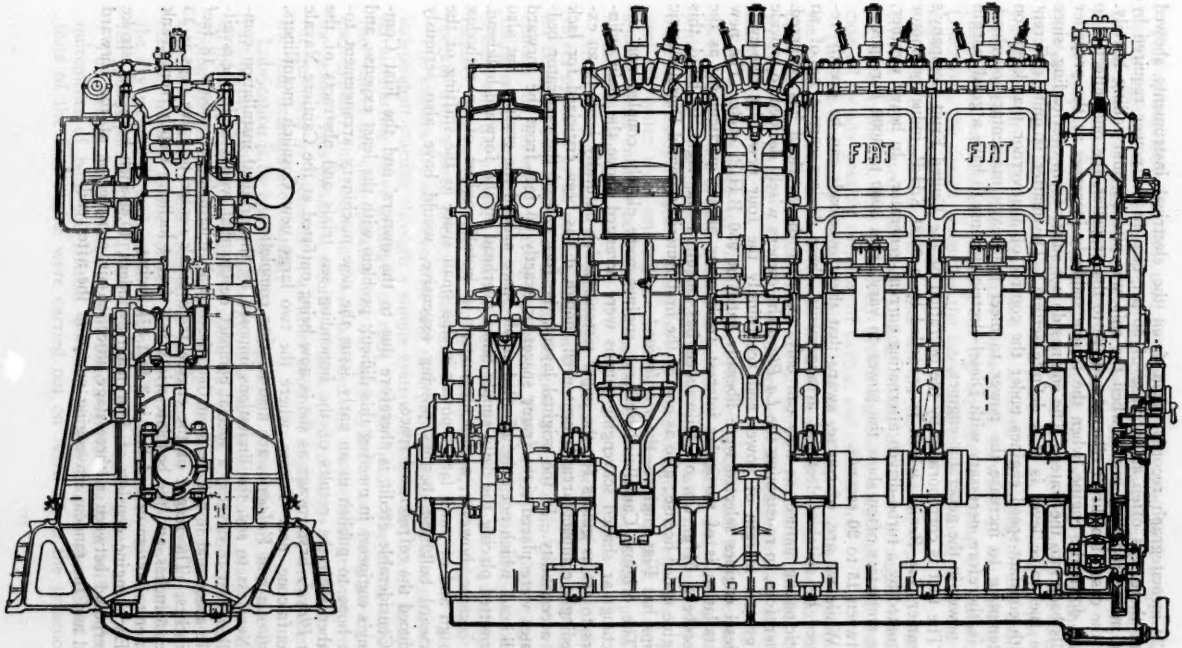
It will be remembered that the *La Playa* was originally equipped with four 825 B.H.P. at 250 R.P.M. Camellaird-Fullagar two-cycle, opposed-piston Diesel engines, each directly connected to a 500-kilowatt main generator and a 200-kilowatt generator, both being direct current 220 volt, operated at 250 R.P.M. The main electric propelling motor was originally rated 2500 S.H.P. at 100 R.P.M., and is installed in a separate water-tight compartment at the after end. The vessel is extensively equipped with electric driven deck and engine-room auxiliaries, including two carbon dioxide refrigerating compressors.

The four main generators are connected in series on the Ward-Leonard system whereby the voltage is increased, as the current flows through each generator, up to a maximum of about 880 volts. .Speed regulation is obtained by varying the field excitation, current for this purpose being furnished by one or more auxiliary generators which are all connected in parallel. At reduced power one or more main unit is cut out as may be desired.

The main consideration which led to the adoption of the Diesel-electric machinery in these vessels was the large refrigerating and auxiliary load amounting to approximately 400 kilowatts or nearly 22 per cent of the power required for propulsion, which necessitates operating one main unit in port at about 75 per cent of its rated power when the cargo holds are being cooled prior to loading. It may also be mentioned that by placing the switchboards and electric control on a flat above the generators and by super imposing the refrigerating machinery on a second flat above the switchboards, the length of the engine room was materially reduced and the cubic capacity of the vessel increased by some 32,000 cubic feet above what might have been obtained with a direct Diesel drive.

It may also be mentioned that the air coolers for refrigerating the forward holds are located on the cargo side adjacent to the forward engine-room bulkhead which is also heavily insulated.

The experience gained during the construction and operation of the *La Playa* and *La Marea* was presented last November before the Institution of Engineers and Shipbuilders in Scotland by D. N. Shannon, the designer of the original engines, in a paper entitled "Marine Oil Engines," from which we quote as follows:



CROSS SECTIONS OF "LA PLAYA'S" NEW CROSSHEAD DIESEL ENGINES.

"Cinematograph records taken from the electrical instruments showed that there was often an increase of 50 per cent in the power required by the main propelling motor when the ship was pitching and rolling heavily. In the case of a direct oil-engine drive to the propeller, the only effect is to slow down the engine, when the torque suddenly increases as the propeller falls back into the water. The engines do not suffer from overloading, since the fuel-valve lever is at a definite setting. The matter is quite different with constant-speed engines under the control of a governor that keeps on attempting to increase the power to meet the heavy momentary overload. It is, therefore, necessary with Diesel-electric systems to have a good margin of power in the generator engines."

The fact is corroborated by the operation of the United Fruit Company's steamer *San Benito*, which is propelled by a 3000 S.H.P. induction motor actuated by a turbo-driven alternating current generator. In heavy weather, the ammeters often show the power to vary from 3000 to 5000 horsepower between 15 to 20 seconds.

While we are, of course, aware that these overloads can be greatly reduced by suitable electrical arrangements, these devices cannot be of an anticipating nature, as they can only act after the overload has occurred. Therefore, in re-engining the *La Playa* the owners wisely decided to provide a greater margin of power and accordingly the four new Fiat two-cycle Diesel engines selected were liberally rated at 950 B.H.P. each. The new installation is of particular interest from many angles, particularly as the Nordberg Mfg. Co. of Milwaukee has acquired the right to build this engine under license; also as one of the first American-owned Diesel-electric ships has Fiat engines—the *Fordonian!*

The original Camellaird-Fullagar engines were closely coupled and the rectangular shaped scavenging pumps were arranged above the main cylinders to act as guides for the upper pistons, which naturally leads to an exceedingly compact arrangement even though it has been criticized for lack of accessibility. In the original layout, the fore end of the generator bedplates were placed eight frame spaces or exactly 20 feet from the forward bulkhead, which not only afforded ample space around the engines, but also permitted placing the daily supply tanks overhead on the forward bulkhead. It became, however, a difficult matter to install engines of an orthodox design and of much larger power in this small space as the shifting of the forward bulkhead, besides proving expensive, would have also seriously reduced the refrigerated space.

Considerable credit is therefore due to the owners' and the Fiat Company's engineers in meeting this difficult problem with the least expense, and we hope to publish in an early issue the new machinery arrangement together with the results of the impending sea trials and abstracts of the *La Playa's* first voyage as she is now being outfitted at the Cantiere Navale Monfalcone, near Trieste, where the two large new Cosulich motorliners, *Saturnia* and *Vulcania*, are also nearing completion.

Needless to say, the limitations imposed by the original installation compelled Fiat to develop a special compact design that would fit in the available space and it will be seen that the overall length is only about 18 feet 6 inches. The height of the engine from crank center to valve tops is 13 feet 5 inches, and the bedplate width is 8 feet 4 inches. Depth from crank center to bottom of bedplate is 3 feet 8⅞ inches.

Each engine consists of four main cylinders, 19.69-inch bore and stroke interposed between a three-stage injection-air compressor at the forward end and the tandem scavenging pumps at the after end.

This arrangement was first used by Fiat on high-speed light-weight locomotive engines built for the Italian and Swedish railways, and was adopted for *La Playa* for compactness and to improve the balancing of the moving parts with consequent reduction of vibrations.

The bedplate was proportioned so as to fit the existing foundations and the forward end of the generator bedplates; in order to minimize the length, the crank shaft has been forged in one piece with an overhanging portion forward to drive the compressor and the attached lubricating oil and cooling water pumps. The space available immediately forward of the scavenging pump crank was cleverly utilized for the skew gears driving the camshaft. Otherwise, the design resembles the latest and most improved Fiat marine engines of much greater power.

It will be noted that the engines are of the crosshead type and that a division plate, fitted with deep stuffing-boxes, has been arranged beneath the main cylinders, thus effectively preventing the contamination of crankcase oil by any possible dripping of carbon and cylinder oil.

The cast-iron liners are in one piece, held at the top by the studs retaining the cylinder heads, and are free to expand downward. Cast steel has been used for the cylinder heads. They are of simple symmetrical conical construction, and project well inside the liners to enhance the cooling of their thickest part.

Salt water is used for cooling the cylinder jackets and large inspection doors are provided at their bottom to permit of inspection and washing down with a hose. The pistons are also salt-water cooled with the usual arrangement of telescopic pipes. These, however, are designed to effectively prevent salt water from entering the crankcase, as they are packed by outside stuffing-boxes entirely accessible through the liberal sized openings provided in the entablature.

Air is distributed to the scavenging pumps by means of tandem piston valves operated by an eccentric mounted on the crank shaft, the scavenging air being led to a large receiver, cast integral with the main cylinders, which contain the automatic scavenging valves. These valves are plain rings of stainless steel which lift as soon as the air ports are uncovered by the main pistons and the pressure within the cylinder is less than the scavenging-air receiver. Being light and of multiple construction, these rings naturally operate with a small pressure differential.

The engines are non-reversible and each pair of cylinders is arranged for independent starting through a simple system of shafts and levers actuating an air-starting valve arranged on each cylinder head. Therefore, only the fuel valves are cam-operated, one for each cylinder, and being co-axial to each cylinder, this results in a well-balanced and symmetrical arrangement of the heads. This minimum number of cam-operated valves should also make for quietness of operation in a rather crowded engine room, even when all the four engines are running.

Lubrication is by big gear-type pump, which draws the oil from the lower part of the crankcase and discharges to a tubular oil cooler, neatly arranged on the compressor frame directly above this pump. As will be seen from the photograph of the engine, the piston and jacket cooling pumps, together with their air chambers and controlling valves, are also arranged in a neat and compact manner at the forward end of the engine. We are informed that the dry weight of each engine complete is approximately 80 tons or about 180 pounds per B.H.P. This is low for a crosshead engine.

The owners having specified that these engines should be built to the requirements of both the British Corporation and Lloyds, the official shop tests of the first engine were carried out on April 15 in the presence of

Captain D. H. Young, superintendent engineer, United Fruit Company; F. G. Knott, of Messrs. Esplen & Sons, who inspected the construction of these engines; and representatives of Lloyds and the British Corporation.

Official Shop Test—Engine No. 1401—Type "Q-504"

M. S. LA PLAYA

April 15, 1927

Test	No Load	½ Load	¾ Load	Full Load	Overload
Duration, hr.	1	2	2	12	4
R.p.m.	257.5	257	253	252	272.5
Load on brake, lb.	———	1069	1598	2138	2204
B.hp.	———	483	728	965	1075
Fuel lb., per b.hp-hr.	———	.448	.414	.405	.442
Average m.i.p., lb./sq. in.	18.37	51.45	61.3	79.15	81.53
Total i.h.p.	281	800	940	1210	1346
Mechanical efficiency	———	.604	.774	.796	.80
Injection air, lb./sq. in.	568	782	853	1066	1066
Lub. oil and cooler inlet, lb./sq. in.	38	39	38	40	40
Lub. oil and cooler outlet, lb./sq. in.	29	31	31	31	31
Cooling water to jacket, lb./sq. in.	17	16	15	17	17
Cooling water to pistons, lb./sq. in	11	12	11	13	14
Scavenging air, lb./sq. in.	2.47	2.22	2.32	2.38	2.75
Cooling water inlet, deg. Fahr.	58	58	58	58	58
Returns from jackets, deg. Fahr.	85	72.5	79.5	84	86
Returns from pistons, deg. Fahr.	93	76	82	89	92
Lub. oil before cooler, deg. Fahr.	102	95	101	101	112
Lub. oil after cooler, deg. Fahr.	80	77	80	84	88
Aver. ex. tem. at cyls., deg. Fahr.	266	343	449	557	627

Exhaust clear throughout.
Compressor aid suction ¾ open throughout.
Fuel oil—90 specific gravity at 50 deg. Fahr.

Several non-stop trials at fractional powers and overload were run with consumption, as noted in the attached table. The fuel oil used had a specific gravity of 0.90 at 60 degrees F.

Vacuum oil D.T.E. extra heavy was used for the main and compressor cylinders and during the 12-hour run at full load the total consumption was 32.2 pounds. Throughout the series of trials the exhaust was almost clear, only a slight haze being visible and silencing was efficient.

During the no-load period the governor was severely taxed on many occasions from overload to no-load and vice versa, and sensitive graphs showed that at no time the change of revolutions exceeded 3 per cent in each direction. At the termination of the trials the cylinders were immediately opened up and pistons removed and examined, all rings found free in their grooves and markings satisfactory on both pistons and liners, showing that they had been adequately lubricated. Likewise, the bronze gear wheels for the camshaft drives were examined and their markings were found regular. In general, the trials were highly satisfactory to all concerned as may be evidenced by the remarkable high efficiencies and low fuel consumptions obtained which compare most favorably with engines of much larger power and better ratio of stroke to bore.

Advantage was taken of the re-engining to bring the whole installation up to modern motorship practice. The original daily supply tanks attached to the forward bulkhead were removed to provide space for the new Diesel motors and a new two-compartment tank installed in the engine-room casing in the space formerly occupied for the storage of galley coal. An additional gravity tank was also installed higher up to supply two fuel-oil separators which drain directly to the daily tanks located underneath. There previously was installed a De Laval lubricating-oil centrifuge and the two fuel-oil separators were added. Extensive improvements were also effected in the ventilation of the engine room by the addition of exhaust fans to handle the heated air from the main generators and new supply fans discharging to ventilating mains with branches to the lower engine room and the various flats.

With the exception of a thorough cleaning and general overhauling, no changes whatever were made to the main and auxiliary electric equipment, which has performed in a highly satisfactory manner since the vessel has been in operation.

The plans and specifications for this conversion and the general engineering and advisory work were contributed by A. Conti, 11 Broadway, New York, in collaboration with the owner's superintending-engineer, Captain H. H. Young.—"Motorship," September, 1927.

THE *LEXINGTON* AND *SARATOGA*.

Those enormous vessels, *Lexington* and *Saratoga*, now being completed in the States, were originally intended to be battle cruisers, but the Washington Treaty brought about their conversion to aircraft carriers. Their huge size is due to the ideas which dominated their original design, for, although their great length and beam will cause them to be admirable aircraft carriers, it is certain that had they been originally designed for the latter purpose their dimensions and power would have been much more moderate than is the case.

The *Lexington* and *Saratoga* are 888 feet in length, 105 feet in beam, and 74 feet 3 inches to the flying deck. They are the largest ships which have ever been constructed in America, and their launching weight has only been exceeded twice in the history of shipbuilding.

But for the Washington agreement it is certain that even larger vessels than these leviathans would have been under construction, for no other type offered greater potentialities of fighting power than did battle cruisers. Associated with high speed, which was desirable from tactical considerations, the offensive and defensive powers of the line-of-battle ship were incorporated in the general design, so that it is not improbable that the battleship of 21 or 22 knots speed would have given place to vessels of the *Hood* type, which seem to have all the desired qualities reasonably balanced. Such characteristics as are required can only be had with ships of enormous size, for each demand for heavier armament, better protection, and higher speed has in it the seeds of growth.

That this is the case is well illustrated by a comparison of the dimensions of the ships of the type which followed their inception over twenty years ago.

In 1905, Japan laid down two cruisers, the *Ikoma* and *Tsukuba*, which were 440 feet in length, 13,750 tons displacement, and of 21 knots speed. They carried an armament of four 12-inch guns mounted in pairs in two barbettes, one forward and one aft, twelve 6-inch guns in casemates, and twelve 4.7-inch guns. They were protected by a complete Krupp steel belt, 7 inches to 4 inches in thickness, with 7 inches of armor on the barbettes. They were followed by the *Kurama* and *Ibuki* a year later, which were 450 feet in length and 14,600 tons in displacement, the twelve 6-inch guns of the *Tsukuba* type being replaced by eight 8-inch guns mounted in pairs in barbettes and two more 4.7-inch guns added. The *Ibuki* was fitted with turbine machinery of 27,000 H.P., the *Kurama* with reciprocating engines of 22,500 I.H.P. The mounting of heavy guns in ships of this class had not been previously adopted and the advantages which this combination of speed and fighting power offered were appreciated by naval constructors all the world over.

In 1906, the British Admiralty laid down three armored battle cruisers, the *Invincible*, *Inflexible*, and *Indomitable*, which, of course, were a great advance on the Japanese vessels referred to above. These ships were 530 feet in length, of 17,250 tons displacement, and were armed with eight 12-inch guns mounted in pairs in four barbettes.

Sixteen 4-inch guns were also carried, and a protective belt of armor 6 inches thick amidships tapering to 4 inches at the ends was afforded. A speed of 25 knots was obtained, the designed horsepower being 41,000.

A steady advance in size was distinctive of the type, and their growth is indicated by the following:—

——	Date.	Length B.P.	Breadth.	Draught.	Displ.	Speed.	Power.
		Ft.	Ft.	Ft.	Tons.	Knots.	S.H.P.
Invincible ..	1907	530	78½	26	17,250	25	41,000
Indefatigable ..	1909	555	80	26½	18,750	25	43,000
Lion	1910	660	88½	28	26,350	28	70,000
Tiger . ..	1912	660	90½	28½	28,500	30	108,000
Repulse ..	1915	750	90	25½	26,500	32	112,000
Hood	1918	810	104	28½	41,200	32	144,000*
Lexington .	1925	850	105	30	45,000	34	180,000

* On trial the *Hood* developed 160,000 S.H.P

As battle cruisers, the *Lexington* and *Saratoga* would have a draught of about 30 feet; but, of course, the displacement of these vessels as aircraft carriers will be much less. The weights allowed for armament and protection would amount perhaps to about 20,000 tons. In the *Hood* the displacement at the load draught was 41,200 tons, of which 33½ per cent was allocated to armor and protection and 12½ per cent to armament, making altogether 19,000 tons. On the other hand, the launching weight of the *Lexington* has been given as 25,500 tons, which would suggest that the finished weight would be around 28,000 tons. To this must be added about 6,000 tons for machinery, giving an approximate light weight of 34,000 tons.

The most striking feature in the above figures is the enormous increase in the power of the propelling machinery. From the time of the *Invincible* class the horsepower has been more than quadrupled, and, if we assume that the *Lexington* will achieve a speed of 34 knots, this has resulted in an increase in speed of 9 knots.

Although the building of these large battle cruisers was a matter of great interest to naval constructors and of pleasure, mixed with some profit and not a little anxiety, to the shipbuilder, their enormous cost made it almost compulsory that some arrangement should be come to whereby the building of them should cease. As battle cruisers the *Lexington* and *Saratoga* would cost in the region of ten million pounds sterling each, and were competition to take place between the maritime powers as to which would possess the greatest number of the most powerful vessels of the type, it would be a serious matter so far as taxpayers were concerned.

These great vessels are notable as being among the greatest achievements of man, leaving, however, no enduring record. In two decades at the most the best of our present-day leviathans will be but a memory, to be succeeded, perhaps, by still larger and faster vessels, if the limit has not yet been reached in the size, which can possibly be built.—"Shipbuilding and Shipping Record," September 8, 1927.

A NEW TYPE OF STEAM ENGINE.

By Dr. Ing. Carl Commentz.

The engine with which this article deals is of the reciprocating type, and is noteworthy on account of the very low values of the fuel and steam consumption figures obtained.

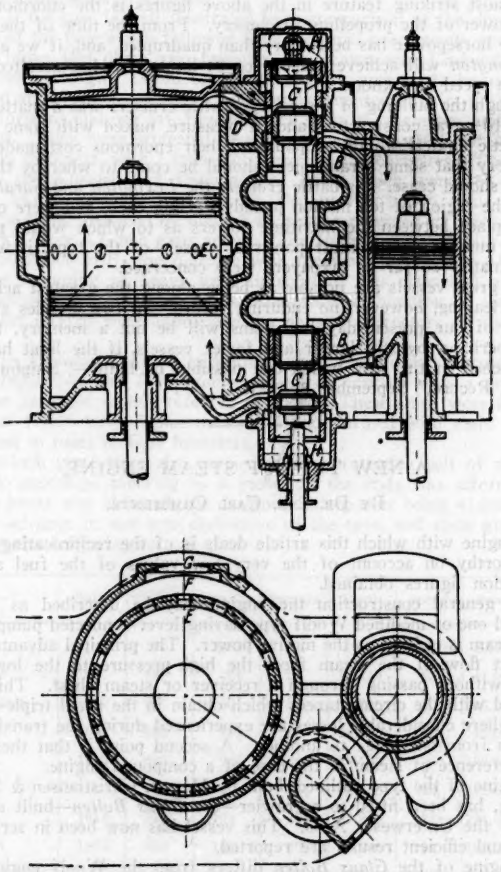
In its general construction the engine may be described as a double-compound one of modified Woolf type having lever-connected pumps. Superheated steam is utilized as the motive power. The principal advantage lies in the direct flow of the steam from the high-pressure to the low-pressure cylinder without passing through a receiver or steam chest. This may be contrasted with the circumstances which obtain in the usual triple-expansion engine, where considerable losses are experienced during the transference of the steam from one stage to another. A second point is that there is only one transference of steam in the case of a compound engine.

An engine of the type outlined, built by Messrs. Christiansen & Meyer, of Hamburg, has been fitted to a trawler—the *Claus Bolten*—built at Geestemunde by the Unterweser A.-G. This vessel has now been in service some months, and efficient results are reported.

The engine of the *Claus Bolten* differs from the Woolf engine in two important details. There is only one reciprocating part governing the steam

transference, *viz.*, a piston valve between the high-pressure and low-pressure cylinders. The second innovation is that the low-pressure cylinder exhaust is by a series of valve ports arranged around the cylinder at half height.

The details of the path followed by the steam may be gathered from the accompanying drawing. It enters at A into the distribution space of the piston valve and through the passage B into the high-pressure cylinder, where it expands until the piston is in its lowest position. The steam then passes back through the passage B into the space C, and through the passage D into the low-pressure cylinder, the piston of which is set at a crank angle



THE MODIFIED WOOLF STEAM ENGINE.

of 180 degrees to the high-pressure piston. When the low-pressure piston is almost in its extreme position, the steam flows through the exhaust ports E into the peripheral passage F, and thence to the condenser. On the return stroke of the low-pressure piston, there is a certain amount of steam trapped in the low-pressure cylinder. If this were to remain, power would be absorbed in effecting its compression. In addition, therefore, to the peripheral ports E, arrangements are made so that the steam may exhaust through C into the space H, and thence to the condenser.

The economy of the engine is to be found in the small cooling surfaces of the passages between the two cylinders, and in the fact that the low-pressure cylinder is operating on the unit-flow principle, which has very decided advantages. The cooling surface of the detrimental spaces amount to only 63 per cent of the total cylinder surface in the high-pressure cylinder, and to 60 per cent in the case of the low-pressure cylinder. These give rise to a very small entrance condensation. The economy of steam engines is known to be very sensitive to such factors.

The other parts of the engine are of normal design and construction. The high-pressure cylinder has a special liner. The engine consists of two units of the type described mounted on the same bedplate, the cranks of the two engines being arranged at an angle of 90 degrees. Lubrication is by mechanical means. The thrust bearing is of the single-collar type. As there is only one steam distribution chamber for each part of the engine, and as this is moreover arranged slightly to one side, the length of the engine is quite small in spite of the four cranks.

The dimensions of the cylinders for the *Claus Bolten* are 315 millimeters and 360 millimeters diameter, by 660 millimeters stroke (12.4 inches and 14.2 inches diameter, by 26-inch stroke), and the engine is capable of developing 525 indicated horsepower at 106 R.P.M., the boiler pressure being 14 atmospheres (200 pounds per square inch). The heating surface is 165 square meters (1775 square feet), the boilers having natural draught. The grate area is 4.6 square meters (50 square feet), and the superheating area 68 square meters (731 square feet).

On a six-hour trial trip, the fuel consumption corresponding to 525 H.P. amounted to only .52 kilogram per I.H.P.-hour (or 1.15 pounds in English units), the coal being good Westphalian. The vacuum was 84 per cent, and the feed-water was heated to a temperature of 75 degrees C. (167 degrees F.).

On a speed trial over a measured course the engine developed an indicated horsepower of 673 at 115 R.P.M., a speed of 11.5 knots being attained.

In considering these results, it must be recollected that the engine is of very small dimensions and that larger plants should show still more favorably, especially if forced draught and other special measures for economy are applied.

It is intended to build the engine to a standardized scheme, including units of indicated horsepowers varying from 400 to 2000. The revolutions will vary from 110 in the smallest to 70 in the largest installations.

The operation of the engine in the trawler *Claus Bolten* has proved to be very simple, and its quick starting under full load has been found of special advantage when setting out the nets and boards.

The fuel consumption realized on the trials has been confirmed by the subsequent fishing trips. In this connection, it may be mentioned that the *Claus Bolten*, of which the bunker capacity is 210 tons, may steam a distance of 8000 to 8400 miles, remaining at sea for so long as 34 to 36 days, without the need of rebunkering.—"The Shipbuilder," June, 1927.

NEW BESSEMER DIESEL HAS MANY REFINEMENTS.

Eight-Cylinder, En-bloc, Engines Closely Follow Automotive Practice.

Unusual freedom from vibration and a noteworthy simplicity of design characterizes the new Model MR-8 Bessemer 1500 horsepower oil engine recently completed by the Bessemer Gas Engine Company. With a bore of 18 inches and a stroke of 22 inches operating at a normal speed of 300 R.P.M., these engines are making a record on the test floor which is a significant commentary on the performance which may be expected of them when put in service in the various types of industrial and marine installations for which they are destined.

As shown in the accompanying illustrations the general lines of the units are strongly reminiscent of automotive practice and this impression is further carried out when a detailed study is made of the structural details. The present description of the MR-8 model includes detailed reference to several features which are found only on the marine types, but these features in some cases suggest control and auxiliary refinements which may be of interest to the discerning stationary Diesel plant operator.

Chief among these is the simplicity of the control gear which is centrally located and will, with the elimination of the reversing mechanism, be even more simplified in stationary practice. There is also a high degree of accessibility, as any individual cylinder head can be removed with very little necessity for dismantling the cooling-water connections or valve-operating gear.

The complete enclosure of the valve mechanism, in a manner which suggests the Italian automobile engines, is another unusual expedient which will add greatly to the appearance of the engine and give an additional factor of protection not ordinarily found on engines of this size.

DETAILED DESCRIPTION OF ENGINE.

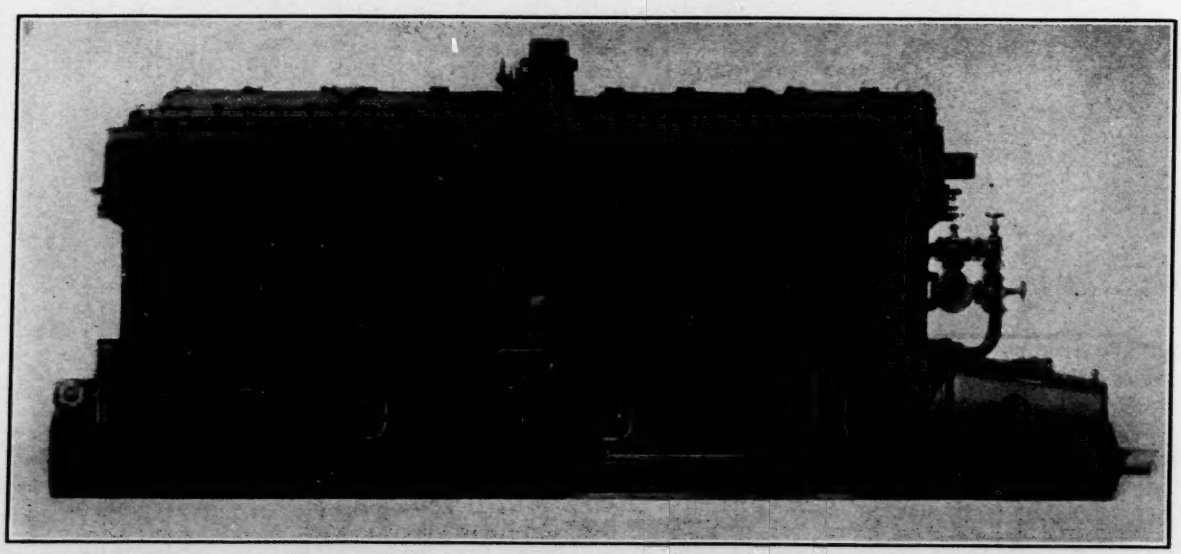
The base of the engine is built in two sections of four cylinders each. All of the base castings and the other castings on the engine are made either of steel or of semi-steel, a feature which adds greatly to the torsional rigidity of the engine and has also been a feature in reducing the over-all dimensions of the unit.

On the forward end of the base is mounted a barring-over-device consisting of a pinion on the crank shaft operated by a worm gear, which in turn can be moved from the outside by a large ratchet wrench. This hand-operated barring gear can also be supplemented with a motor drive if desired. At the rear of the engine is located the fly-wheel and overspeed governor, and in the marine type a Kingsbury thrust bearing is also installed.

CRANK SHAFT.

The crank shaft is forged in two sections, each serving four cylinders. This shaft, which embodies the results of considerable investigation by H. F. Shepherd, Chief Engineer of the Bessemer Company, is built to very generous dimensions,—the diameter of the main bearings and the crank bearings being 12¾ inches.

In a series of exhaustive tests it has been found that there are no critical speeds up to 350 R.P.M. which is rather remarkable for an engine of this size. It is therefore not surprising that at the normal speed of 300 R.P.M. the operation of the engine is as quiet and as relatively free from vibration



THE BESSEMER, TRUNK-PISTON, EN-BLOC TYPE, MR-8 DIESEL.

as the most delicately balanced eight-cylinder automobile engine, a type which it closely resembles. This freedom from vibration of the engines on the test floor is particularly impressive when one notes that the engine beds are merely supported on cast-iron laterals, rather than being bolted to their mounting in the usual way.

At the center of the crank shaft coupling in the middle of the engine is mounted a pinion which operates the chain drive to the overhead cam shaft through a 2-to-1 reduction gear. The location of this cam shaft drive in the center of the engine has undoubtedly been a factor in achieving the steadiness of operation and compactness mentioned above and it also possesses the additional advantage of permitting the control station to be located at the middle of each unit.

CENTER FRAME.

The two center frame housings or water boxes each enclosing four cylinders are of "H" type construction exceptionally well braced. The cylinder liners are of semi-steel and are bored and ground to a smooth finish. A copper-asbestos gasket is fitted at the top of the liner, and at the bottom is a rubber gasket fitted with a removable gland for sealing the water circulation space in the center frame. Eight large cleanout plates are located at equal intervals along the center frame. Above these plates are located the lubricating oil leads through which lubricating oil is furnished to the cylinder walls by special Manzel timed lubricators.

PISTONS.

The pistons are made of Bu-nite nickel alloy with four steel bands cast internally to provide a guide for the expansion of the heated piston metal.

The wrist pin bearings are 8⅞ inches in diameter and the connecting rods are of forged steel machined all over. Bearings are of cast steel with removable shells which are babbitted by a centrifugal process.

CYLINDER HEADS.

It will be noted that an excellent distribution of material has been obtained by the use of dual valves, greatly reducing the liability of thin sections in the cylinder head castings. The heads are also baffled internally to assist the distribution of the cooling water and relatively large cleanout plates are provided for access to the water space. The circulating water is taken from the jackets in the head to the jackets surrounding the cylinder liners by passages located at four points equi-distant around the circumference of the head. Their simplicity and careful proportioning to resist undue heat stresses is quite apparent.

CAM SHAFT AND VALVE GEAR.

The overhead cam shaft is supported by large brackets bolted to the cylinder frame and attached to each of these brackets is an additional cast upright which supports the rocker arm shaft. Each pair of valves is operated by a single rocker arm which transmits its impulse to them through a hinged bridge. This method of transmitting the impulse from the rocker arm to each pair of valves constitutes a distinctly original solution of the dual valve problem.

The fuel injection valve, located on the axis of each cylinder head, is operated by an inclined push rod which rides directly on the cam shaft. An auxiliary shaft which is mounted parallel to the cam shaft controls the movement of this push rod when the overspeed governor is brought into play. The governor gear is an excellent example of the growing application of ball bearings to oil engines, as the linkage and shafts are entirely supported on bearings of this type.

An air starting pilot valve on each cylinder head is also operated by the cam shaft. These pilot valves open the main air starting valves to each cylinder at the proper time for starting and are automatically closed when the engines begin to operate on fuel. The relief valves are mounted as usual on the side of each cylinder head.

For reversing in marine service the cam shaft is mounted on eccentric bearings, the rotation of which permit the shaft to be disengaged from the rocker arms and moved laterally in order to bring the reversing cams into position.

Movement of the cam shaft is carried out by an air ram mounted in the center of the engine in the same plane as the chain and pinions which operate the cam shaft. The piston of the air ram is direct connected to a rack which rotates the cam shaft and also brings into play a grooved cam, supplying the impulse for bringing the reversing cams into position.

The air starting valves, the air ram described above and the fuel pumps are all controlled by a lever which works in a "U" shaped slot mounted at the top of the control housing. At the base of the "U" the lever is in the "stop" position. Moving it up in either limb of the "U" opens the air starting valve. As the lever is moved up the slot, the volume of starting air is gradually increased and the fuel pumps are then brought into play as the control lever progresses through the full length of its swing. During this operation an additional factor of safety is provided by a lock on the control lever which must be depressed before it passes to the point where fuel is admitted.

Reversing the movement of the lever gradually cuts off the supply of fuel and the lever is shifted to the opposite limb of the "U", where a similar air starting impulse is given in the reverse direction, this being followed with an increase in the fuel supply in exactly the same manner as described above.

PUMPS.

The fuel pumps, mounted directly on the center frame, are operated by a pinion engaging with the cam shaft drive. All other auxiliary pumps and compressors are operated as independent units.

The circulating pump for the water cooling system is a DeLaval unit direct connected to a 7½ horsepower Westinghouse motor. The lubricating oil pump is a Northern 4-vane type of 120 gallons per minute capacity. It is likewise driven by a 7½ horsepower motor mounted on a unit sub-base.

A Schutte-Koerting oil cooler and an oil strainer of similar make is also mounted on the flywheel end of the engine.

The auxiliary compressors for furnishing maneuvering and starting air are Ingersoll-Rand 2-stage 21 cubic feet capacity units, also operated by motor drive.

For marine installation, auxiliary generating equipment, consisting of two 6-cylinder model F-6 Bessemer Diesels direct connected to 110 kilowatt

Westinghouse generators are supplied, in addition to one type E-3 3-cylinder Bessemer Diesel rated at 60 horsepower direct connected to a 40-kilowatt Westinghouse generator.

The marine practice followed in installing these engines also calls for water pumps and lubricating oil pumps which are of an individual capacity large enough to serve two engines in the event that one of the units breaks down. This innovation was primarily adopted for marine service but for large stationary plants, utilizing two or more units, it seems to be a consideration well worthy of the attention of the industrial Diesel operator.

The magnitude of the facilities which are available in both the machine shop and the erecting shop is quite apparent. The operations of the Diesel division of the Bessemer Company are under the personal supervision of Dr. Fithian, president and co-founder of the company; H. F. Shepherd, Diesel engineer, and C. M. Reagle, Diesel sales manager.—"Oil Engine Power," October, 1927.

WHEN DOES A BEARING POUND IN AN INTERNAL COMBUSTION ENGINE?

By L. H. Morrison.

This is a discussion of bearing pressures, intended to show that current ideas as to when a bearing pounds are erroneous. While the Otto cycle gas engine is used as an illustration, the reasoning applies to all gas engines.

All who have operated oil engines, or for that matter any reciprocating engine, know that if the connecting-rod bearings are given too much clearance they will pound. But just what makes them pound and at what point in the cycle the pounding occurs is not so clear.

Discussing the wristpin only, at first, most operators will say that with a four-stroke-cycle single-acting engine no pound can occur on the power stroke since the wristpin is in contact with the one side of the bearing during the entire stroke and on the following exhaust stroke the piston is forced back with the pin still pressing against the same side of the bearing. It is at the end of the exhaust and beginning of the suction stroke that the pound occurs, so many assume, since at this point the pin leaves its contact with the bearing side to make contact with the opposite side of the bearing.

The assumed conditions are shown in Figure 1. A piston position is shown for each of the four strokes with the pin in contact with either the bearing side L or R. In addition, in the center has been placed a typical gas-engine indicator diagram showing the lines that would be drawn by the indicator pencil during the four strokes.

If these pressures be laid out along a line, we obtain a pressure curve for the four strokes, as shown in Figure 2. For convenience the exhaust and suction strokes are assumed to be atmospheric, the error by so doing being negligible. The four strokes are marked R or L to indicate which half of the bearing is in contact with the wristpin.

On the face of it the bearing should pound when the direction of bearing pressure apparently changes at the end of the suction stroke. But this is not the state of affairs, as another factor creeps in to destroy the entire argument. This factor is inertia.

It requires power to start and bring an automobile up to speed, and power, in the shape of braking action, is developed in slowing it down. In the same way a force must be exerted to bring an engine piston up to its maximum speed and then to slow it down as it nears the end of its stroke.

This acceleration and slowing down require a force to be exerted. It is obvious that during the early part of the stroke, provided no other force

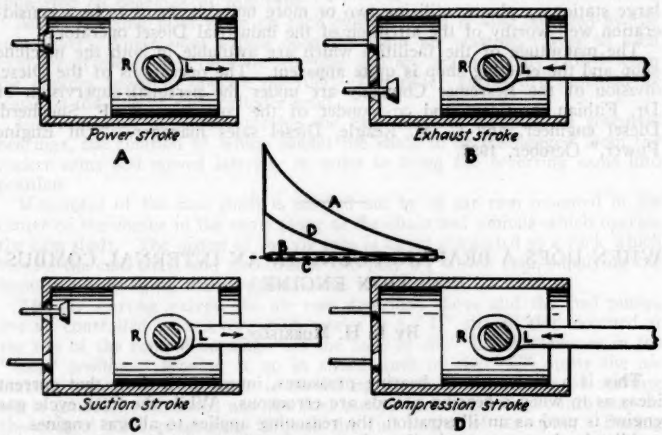


FIG. 1.—POINT OF WRISTPIN CONTACT WITH ITS BEARING OF GAS PRESSURE WAS THE ONLY FORCE PRESENT.

existed save that used to perform the acceleration process, the crank and rod would pull the piston and during the slowing down process the piston would exert a force on the connecting rod and wristpin. The force necessary to overcome the inertia of the piston in acceleration can be calculated by the equation

$$F = \frac{WV^2}{gr}\left[\left(\cos a + \frac{\cos^2 2a}{n}\right)\right]$$

when a = crank angle; r = crank throw; n = rod to crank ratio; V = velocity of the crankpin; W = weight of reciprocating parts; and g = acceleration due to gravity, feet per second per second. Using this formula and inserting the values pertaining to a particular engine, the forces neces-

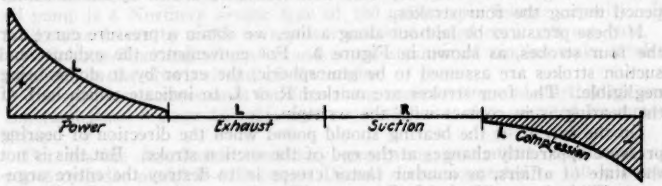


FIG. 2.—DIAGRAM SHOWS THE PRESSURE CHANGES DURING THE CYCLE.

sary to overcome the inertia at all points in the stroke can be calculated and laid out as shown in Figure 3. At that part of the stroke where the inertia force curve is below the base line, the force is negative—that is, it resists the piston motion—and when above the line, the inertia force assists the continued piston travel. The curve has been constructed for all four strokes, and marked R and L to indicate which side of the bearing the wristpin is in contact with if there were no gas pressure on the piston. Assuming that no other force is present, such as when the engine is running idle, it will be noticed from Figure 3 that the pressure on the bearing reverses at approximately mid-position in each stroke. The distance the curve is above or below the base line is proportional to the intensity of the force in pounds per square inch of piston area, in order to make it to the same scale as the indicator diagram in Figures 1 and 2.

If we combined Figures 2 and 3, we would obtain the net force tending to move the piston on its stroke. For example, during the early part of the power stroke the force necessary to overcome the piston inertia resists the force due to the gas expansion, and as a consequence the net force tending to turn the crank through the agency of the connecting rod is diminished, and during the remainder of the stroke the net power is increased by the inertia force. A combination for the four strokes gives the net force diagram shown in Figure 4.

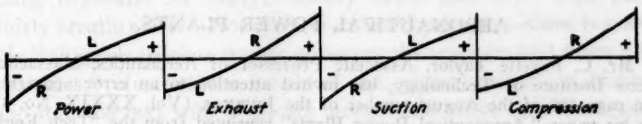


FIG. 3.—THE INERTIA OF THE RECIPROCATING PARTS RESULTS IN THE INTRODUCTION OF ANOTHER FORCE.

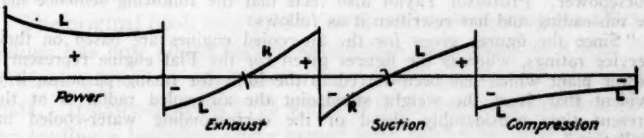


FIG. 4.—THE COMBINED GAS AND INERTIA FORCES GIVE THIS NET-FORCE DIAGRAM.

It will be seen that during the entire power stroke a positive force is applied to the rod and the pin is always in contact with the left side of the bearing. During the exhaust stroke the net force is negative during the first portion and positive during the second portion of the stroke. The pressure of the wristpin changes from the left to right side of the bearing at about the center of the stroke, but the change is so gradual that unless the clearance be excessive no pounding will occur. The same applies to the suction stroke.

During the compression stroke the mean-effective pressure is at all times negative, and at no point in the stroke does the positive inertia force equal the negative gas pressure. Consequently, the net force curve is negative during the entire stroke, with the pin in contact with the left side of the bearing.

On the next power stroke the pin pressure remains at the left, so that no reversal occurs at the end of the compression stroke, but the value of the total force is increased at dead center, owing to the combustion of the gases.

It is clear, then, that the wristpin cannot pound at the ends of any stroke. It can do so near the center of the suction and exhaust strokes, but only if the clearance is large.

A lighter piston would cause the inertia force curve to be flatter, resulting in a lesser pressure change during these two strokes. Clearances with light-weight pistons can then be greater. Conversely, a heavier piston or a higher speed would make the curves steeper and the more rapid pressure change would require closer clearances to avoid pounding. To offset high speeds, light pistons are necessary.

It is interesting to note that if a gas engine having heavy reciprocating parts should operate on a low load or have a leaking piston, the net-force diagram during the compression stroke may have a different shape from that shown. Instead of a negative value during the entire stroke, the curve may cross the zero line near the end of the stroke. This would cause pounding at the end of the compression stroke. In one case the curve crossed the zero line twice in the stroke, producing two distinct thumps.—"Power," October 11, 1927.

AERONAUTICAL POWER PLANTS.

Mr. C. Fayette Taylor, Associate Professor of Aeronautics, Massachusetts Institute of Technology, has invited attention to an error appearing on page 586 of the August number of the JOURNAL (Vol. XXXIX, No. 3) in his paper "Aeronautical Power Plants" reprinted from the "Tech Engineering News."

The weight per horsepower as given for the Fiat engine in the first paragraph should be 1.04 pounds per horsepower instead of 1.03 pounds per horsepower. Professor Taylor also feels that the following sentence may be misleading and has rewritten it as follows:

"Since the figures given for the air-cooled engines are based on their service ratings, whereas the figures given for the Fiat engine represent a power plant which has been forced to the limit for racing purposes, it is evident that from the weight standpoint the air-cooled radial is at the present time considerably ahead of the corresponding water-cooled installation."

BOOK REVIEW.

A SHORT HISTORY OF THE UNITED STATES NAVY, by REAR ADMIRAL GEORGE R. CLARK; PROF. WM. O. STEVENS; PROF. CARROL S. ALDEN; and PROF. HERMAN F. KRAFFT;—Revised and Continued (1927) by CARROL S. ALDEN, PH.D.—560 pages—55 illustrations and maps—$4.50—J. B. Lippincott Co., Philadelphia.

This history is a living story.

It is a shame to brand it as a textbook. That connotes something dry—and the Navys' history never was dry. This book fairly breathes life. Not because of word painting—there is none. No " sluggish crimson stream oozes out the scuppers and flows o'er the sides." Instead, the words of the original makers of history slip into the text with a smooth stream line effect that leaves one feeling he lived through it himself. Nor does the " eagle scream." The words of gallant enemies are quoted as well as those of our own seamen.

The original book was written some seventeen years ago. Prof. Alden has now added such a seamanlike long-splice to it that the present work is a smooth-running cable from the Revolution to date. It extends from 1775 when O'Brien captured our first ship by leading a boarding party armed with pitchforks and axes over the bulwarks of a British schooner, to 1926 when Byrd flew over the North Pole in the latest type aeroplane. It shows that history is still in the making, by mentioning the beginning of the Nationalist uprising in China in the Spring of 1927.

This is a book to keep as well as to read. Its thorough index makes it a facile reference work. As such it would serve all purposes except those of an admiral's staff or a special historian. Even in such cases the well-chosen and arranged list of " authorities" probably would make this book the first to be consulted. Moreover, a succinct analysis of results—sometimes in a sentence, and sometimes in a separate paragraph—ties this special naval his-

tory to the military, political, economic and scientific histories of the country. One is never left with the uneasy query "that was a thrilling battle, but what good did it do?—what of it?"

This is not a "philosophical" history. The authors are not propounding a theory and then citing special facts to prove it. Instead, they have observed the most scrupulous thoroughness and accuracy to present all facts, and the nicest judgment to present them in proper proportion and perspective. Then the reader can form his own theories on a sound basis. This is all the easier because the most significant facts stand out like studding sails on a frigate. Among such things might be cited the following:

The strength of a man-o'-war depends on three things:—accurate heavy hitting power, tough resistance to punishment, and mobility. Again and again the ship and fleet actions illustrate that point.

The strength of our diplomacy varies directly with the strength of our armed forces, and particularly with that of our Navy True, sweet reason should prevail. Usually it does, if only others will listen to it—will attend to it. "What we attend to controls our behavior. What we can get others to attend to controls their behavior." And how nations do "attend" to the reasons of diplomats with a strong Navy on the high seas! And how deaf they are to diplomats without that backing!

Pacifists, like the poor, are always with us. They have done their best to wreck the Navy after every war. Even when ships are authorized by law they try to stop the building. Note the following:—"if a peace should take place between the United States and the regency of Algiers, no farther proceedings shall be had under this act" and "in the event of an international conference for the limitation of naval armaments the President is * * * empowered * * * to suspend * * * construction authorized by this act." The first quotation is from the act of March 27, 1794, for the building of six cruisers (frigates) and the second from the act of December 18, 1924, for the building, among other items, of eight cruisers. No lessons learned in 130 years! These repetitions of history would make a Fundamentalist believe in the transmigration of souls, if only a pacifist had a soul. They lend

color to Weismann's theory of the germ plasm, and inspire a profound regret that in the early days so little was known of vasectomy.

Courageous will power conquers desperate conditions. From Jones' epigram of determination, "I have not yet begun to fight," to the dogged endurance of explorers in dreary Arctic wastes, history demonstrates this time and again. Picture Cushing. Fifteen men in an open boat against the armor-clad *Albemarle* and thousands on shore. He is under heavy fire from rifles ashore and cannon afloat. He dashes his launch over the log barriers, knowing it can never go back. He coolly plants his torpedo and blows up the *Albemarle* "at the same instant that 100 pounds of grape, at ten feet range, crashed among us." He dives into the frost-bitten river and swims most of the night. He crawls in mud and slime all the next day with his enemies nearly stepping on him. By swimming he steals a little skiff from under the noses of seven armed soldiers. He rows for ten hours the next night to report to his Admiral that the ironclad was destroyed and wooden ships could proceed. Then it is recorded that (as the wine mess was not yet abolished) he had a few snifters and took part in the bombardment next day. There was a sailor!

"No great thing cometh suddenly into being." Great successes are the result only of prolonged study, thorough preparation, and energetic work. The public applauds the spectacular success; but only such a history as this shows the "tireless energy, the sound judgment, and the farsightedness that always precede and sometimes follow, a successful battle." Oliver Perry hacked his fleet out of the forest. Farragut never had his opportunity until he was past sixty, yet as a young officer he "had taken great pains to inform myself as to the local advantages in attacking the place, measured the depth of water all around the fort, and marked the penetration of every shell from the French ships." Matthew Perry dredged the bookshops of New York and London for literature on Japan, studied it thoroughly, and was brilliantly successful in opening Japan to Western civilization. "The extraordinary success of Captain Semmes (Confederate Navy) was due to the diligent study he had made of trade routes." Peary reached the Pole after twenty-three years of preparation. "Dewey

has been a life-long student of everything connected with the sea. He is a constant reader. * * * He made a study of harbors. * * * He undoubtedly knew just what he was doing and where he was going when he made that midnight dash which seems to be so amazing to people who don't know him." Officer after officer illustrates this point with his career.

I wish the authors had mentioned the influence of Admiral Sims on the accuracy of our gun fire, when they mentioned the influence of his friend, Admiral Sir Percy Scott, on the accuracy of English gun fire.

Delving into the past is idle curiosity except it be for the purpose of finding a guide or stimulus to present actions leading to a better future. Readers of this history will find such a guide and stimulus.

I hope every educated American reads this book.

LYBRAND SMITH,
Lieutenant Commander, U. S. N.

THE BRIDGE TO FRANCE, by EDWARD N. HURLEY; Published by J. B. Lippincott Co. REVIEWED BY LIEUTENANT COMMANDER G. H. WOOD, U. S. NAVY.

One of the most extraordinary dramas ever connected with the seas was enacted during the late War—the construction and operation of "The Bridge to France." Created in a moment of desperate necessity, the Shipping Board and its war child, the Emergency Fleet Corporation, entered into a program of shipbuilding that was magnificent in its scope and ambition. But this same program, which had seemed so praiseworthy as measured by the efforts and accomplishments of the other allies who had long been under the spur of war stimulation, shrank into insignificance during the early months of 1918 when the backs of the allies were against the wall. As the true situation dawned, as it came to be realized that American troops were needed immediately on the front in an unprecedented number to stem the tide of the German drive, that unless this were done the War would absolutely be lost and the United States be obliged "to pay for the entire cost of the War to the Central Powers," the realization came that upon the

building of ships alone hinged the outcome. At this point occurred the most dramatic act of the whole drama. The troops were ready, the troops ships were available in which they were to be carried over at such an unprecedented rate; but the supply ships that were to prevent them from starving and equip them for fighting were nowhere in sight. "Hurley, We Must Go The Limit" will go down into history as one of the most daring, magnificent gestures of man. This Country and the Allies should ever be grateful that Woodrow Wilson had the foresight to select his leaders as he did, to have faith in them to do the impossible, and to have the sublime courage to risk one of the most appalling military disasters of history.

The book, "The Bridge to France," by the wartime Chairman of the United States Shipping Board, Mr. Edward N. Hurley, is a vivid history of the above. Many of the incidents contained therein have never before been published. "The Launching of Woodrow Wilson," at which Mr. Hurley took a prominent part, into a career of public service was a fitting start to a subsequent brilliant career. The organizing of the Shipping Board and the Emergency Fleet Corporation, the development of the shipbuilding industry of this country to its supreme effort, the creation of Hog Island, the transporting of our troops and supplies to France through the Submarine Zone are all graphically told. This book is a story of an accomplishment, expensive though it was, of which Americans should always be proud.

ASSOCIATION NOTES.

The Annual Meeting of the Society was held at Washington, D. C., on Tuesday, October 4, 1927. The following were nominated for officers for the year 1928:

For *President:*
Rear Admiral John Halligan, Jr., U. S. N.

For *Secretary-Treasurer:*
Commander H. B. Hird, U. S. N.

For *Members of Council:*
Rear Admiral C. W. Dyson, U. S. N., (Retired).
Captain E. L. Bennett, U. S. N.
Captain Q. B. Newman, U. S. C. G.
Captain O. L. Cox, U. S. N.
Commander H. R. Greenlee, U. S. N.
Commander C. A. Dunn, U. S. N.
Commander C. A. Jones, U. S. N.
Lieutenant Commander E. L. Cochrane (CC) U. S. N.
Mr. B. P. Lamberton.
Mr. H. M. Southgate.
Mr. J. F. Metten.

Polls close on December 26, 1927, at 4.30 P. M.

It was voted to hold a banquet in 1928. The President will appoint a Committee to make arrangements, which will be announced in full at the earliest practicable date.

The following have joined the Society since the publication of the last JOURNAL:

CIVIL.

Penniman, Abbott L., Jr., 3405 Greenway, Baltimore, Md.

ASSOCIATE.

Schmidt, Henry F., 85 S. Lansdowne Ave., Lansdowne, Pa.
Scott, Bernard W., 430 Andover St., Lawrence, Mass.